Design of a 2.0 MW wind turbine planetary gearbox

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Abstract

In recent years there has been much attention to the production of electricity from wind; this is performed by wind turbines. They have been used since ancient times to produce mechanical energy with other objectives than producing electricity, like grind grains. But the real explosion, in the use of wind turbines to produce electricity, occurred from 1970 onwards, when, because of the oil crisis, new ways to produce energy were sought.

A central component in every wind turbine design is the gearbox. Its reliability and cost are critical factors in the success of the overall design. Historically, the wind turbine industry has been plagued with gearbox failures, which have affected virtually every type of wind turbine configuration.

Many of these failures can be directly attributed to poor communication between the wind turbine engineer and the gearbox supplier. Successful wind turbine gearbox applications require a close working relationship between these two disciplines, with the active involvement of the wind turbine engineer during the design and procurement process. The wind turbine engineer must have enough knowledge to specify the application, environment, loads, gear life and quality, acceptance criteria, noise, and many other factors, and relate this information to the gearbox vendor. Gearbox life expectancy will depend on the ability of wind turbine operators to make informed decisions about operation and maintenance.

The main objective of this dissertation will be the design and dimensioning of a gearbox capable of adapting the low rotations and high torque of the blades of a wind turbine under conditions of speeds suitable for the generation of energy in the generator. To do so, a research was carried out to evaluate the current state of wind energy production, the type of equipment available, its characteristics and other existing gearboxes. The best performing solution was then chosen for optimization. KISSsoft® 2016 and KISSsys® 2016 were used in the dimensioning of gears, shafts, rolling bearings and keys.

Keywords: Planetary gearbox, Power split stage, Wind turbine, Dimensioning, Design, KISSsoft®, KISSsys®, Lubrication, Oil injection.

To my friends and family ...

‘As engrenagens constituem o verdadeiro ex-libris da Engenharia Mecânica’

Professor Paulo Tavares de Castro

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Contents

List of Figures

List of Tables

Notation

Acronyms

Symbol	Designation
AW	Anti wear
BSP(T)	British standard pipe (taper)
CVT	Continuos variable transmission
DLC	Design load cases
EHL	Elasto-hidrodynamic lubrication
EN	European Standard
EP	Extreme Pressure
EU	European Union
HAWT	Horizontal axis wind turbine
HRC	Rockwell hardness (Scale C)
IEC	International Electrotechnical Commission
ISO	International Organization for Standardization
NPTF	National pipe taper Fuel
NREL	National Renewable Energy Laboratory
PAO	Polyalphaolefin
PE	Polyetylene
PeX	Cross-linked Polyetylene
PL	Planet
PP	Polypropylene
PVC	Polyvinyl Chloride
SN	Sun
US	United States (of America)
VAWT	Vertical axis wind turbine
VG	Viscosity grade
WECS	Wind turbine energy conversion system
WT	Wind turbine
WTG	Wind turbine gearbox

Principal Symbols

Symbol	Designation	Unit
A	Area	mm^2
a	Center distance	mm
b	Facewidth	mm
c	Relief coefficient	μ m
d	Diameter, Reference diameter	mm
f	Frequency	Hz
i	Total transmission ratio	-
$I.D$	Inner diameter of the pipe	mm
k	Heat transfer coefficient	W·m^{-2}·K^{-1}
L	Length	mm
m	Module	mm
N	Angular number of teeth (always associated with an assembly angle)	-
n	Rotational speed	rpm
nb	Number of poles of the generator	-
$O.D$	Outer diameter of the pipe	mm
P	Power	W
p	Pitch	mm
Q	Flow rate	l·s^{-1}
R_a	Aritmethic mean roughness	μm
R_z	Mean roughness height	μm
r	Radius	mm
s	Tooth thickness	mm
T	Temperature	^{o}C
t	Thickness	mm
u	Gear ratio	-
v	Linear speed (velocity)	m·s^{-1}
x	Profile shift coefficient	-
z	Number of teeth	-

Greek Principal Symbols

Symbol	Designation	Unit
α	Pressure angle	^{o}
β	Helix angle	^{o}
γ	Angular pitch	^{o}
Δ	Interval	
δ	Deflection	
Θ	Global angular misalignment	^{o}
θ	Angular misalignment	^{o}
ι	Assembly angle	^{o}
κ	Assembly angle	^{o}
λ	Assembly angle	^{o}
σ	Normal stress	MPa
τ	Shear stress	MPa
ω	Rotational speed	rad·s^{-1}

Principal Subscripts

Subscript	Refers to
a	Tip
b	Base
C	Gearbox surface
e	External
f	Root
i	Internal
L	Dependent of the load
m	Mean
N	Non dependent of the load
n	Normal direction
r	Radial
t	Transverse
w	Operating condition
y	Axial direction

Abbreviated subscripts

Subscript	Refers to
adm	Admissible
gen	Generator
in	Input
INF	Inferior
max	Maximum
min	Minimum
out	Output
rd	Rupture
rot	Rotor (Blades)
SUP	Superior
yd	Yield

Examples of symbols

Symbol	Designation	Unit
d_w	Operating reference diameter	mm
m_n	Normal module	mm
n_{gen}	Rotational speed of generator	rpm
α_{wt}	Transverse operating angle	^{o}

Chapter 1

Introduction

1.1 Background

It is known that energy use grew quickly due to the evolution of technology, becoming its production a huge sector of economies and global industry. However, for much over a century, energy related innovation is mostly fuelled by coal and petrol, causing a remarkable increase in oil price. It is only during the energy crisis of the 1970s and the interruption in Middle Eastern oil exports, that a strong interest in non-fossil power sources arises.

After that, in the last years, a new concern about fossil fuels, related to the impact on the environment, comes up. Burning fossil fuels causes health problems, with air pollution linked to heart and respiratory diseases, and concentration of gases in the Atmosphere, mainly carbon that retains the heat of solar radiation.

Then, with the run-up in oil price, resulted from the perception of energy crisis and the global warming effect, along with the motivation to make the world less dependent on fossil energy sources, led to the search for new ways to produce energy in a proficient and environmentally friendly way. This was found in the exploitation of renewable energy resources; they contaminate less and do not involve the exhaustion of the energy source.

The most important benefits of electricity generation by wind power is their self-sustainability, since for as long as the atmospheric conditions permits, the energy produced can be harnessed to send power across the grid, and it does not produce hazardous wastes, like carbon dioxide emissions, while during operation.

According to [1], renewable energy accounted for 86 % of all new EU power installations in 2016: 21.1 GW of a total 24.5 GW of new power capacity, with wind power representing more or less 51 % of total power capacity of installations. With almost 300 TWh generated in 2016, wind power covered 10.4 % of the EU's electricity demand.

The increase of the needed electrical power, year by year, motivated the production, transport and storage of energy, for usage on demand. Wind turbine is one of the equipments that makes it possible. It may take several configurations, but a rotor connected to a single gearbox plus generator arrangement is quite usual.

In a *gearbox-generator* arrangement, the failure cost comes almost entirely from gearbox collapse, whose repairs require many days of wind turbine (WT) inactivity. So, WTs are fundamentally dependent on a powerful, efficient and reliable gearboxes.

1.2 Objectives

The main goal of this thesis is to design a 2 MW wind turbine planetary gearbox.

Among the different challenges of the design are the geometry of gear teeth and the selection of rolling bearings, in order to maximize the energetic efficiency of the gear transmission. Another challenge is to keep the transmission as simples as possible, in order to control the manufacturing costs, using standard manufacturing technologies, and increase the overall reliability.

For this project, it is assumed the use of KISSsoft® 2016 and KISSsys® 2016 softwares for the main mechanical elements calculations and gearbox layout design, respectively. Notice that, for the decision-making it shall be taken into account the fundamentals of mechanical design.

At last, it is supposed to sketch and draw the final solution among the various configuration which arise. This will be achieved through the 3D CAD design software SolidWorks® 2017-2018.

1.3 Problem definition

Besides the torque from the rotor (or blades), the gear meshing also applies large moments and forces on its own. Then, it is important to ensure that gearbox is designed to support these loads, otherwise its internal components can become severely misaligned and this can lead to stress concentrations and failures. This section is dedicated to clarify some specific problems atypical in this kind of application and to present the design drivers' gearbox

Load Wind turbine gearbox undergo severe fluctuating torques and change of torque direction, during start-ups, shut-downs, emergency stops, and during grid connections; excessive overloads due to vibration; and, loads at standstill. Notice that Load cases that result in torque reversals may be particularly damaging to bearings.

Power Due to the high power level at a very low wind speed, gearbox experience high torques. It is of paramount importance a meticulous selection of rolling bearings in order to minimize power losses.

Operation condition Owing to operating environment, the gearbox faces significant temperature fluctuations. It must be taken into account the cold start with cold lubricant, too.

Dimensions Because of the high height that characterises the horizontal wind turbine, a lightweight design is required. It shall be notice the bending torque on the input shaft (especially for three-point bearing).

Noise As a result of blade motion in air, wind turbine typically presents a high level of aerodynamic noise. In addition to that, mechanical noise, provided by multiple components like gearbox, generator, cooling fans, and others, occurs. So, considering the amount of noise phenomenon that wind turbine is associated, it is important to control this effect during the gear mesh and gearbox housing design and dimensioning.

Accessibility The gearbox is of difficult access, transportation tool is difficult.

Economical High number of damages connected to high costs.

1.4 Layout

This document is organized by Parts and Chapters, constituting the first one an introduction where a brief contextualization and exposition of the proposed problem is presented. A set of references and appendixes can be found at the end.

The **Part I** is devoted to theoretical contextualization and is incorporated by Chapter 2.

Chapter 2: It is concerned with fundamentals of decision-making. It starts to present the state of the art of Wind Turbines, by presenting a description of wind turbines conversion systems, their evolution and global production/consumption, and their main components. Then, it focuses on gearbox utilization in wind turbine drivetrain, by describing it and by introducing different gear shaft arrangements and their market position.

Part II is reserved for the technical development of the work, and includes chapters 3 and 4.

Chapter 3: The fundamentals are applied to select the elements of mechanical systems, and to dimensioning and design the gearbox. Here, the decision-making will be supported, in some cases, by the analysis and comparison of different gear parameters.

After selected the transmission stages number, the definition and dimensioning of the kinematic chain will follow the gear parameters setting, like: number of teeth, module value, gear type, angle propeller, and etc... An analysis of the tooth resistance shall be made, having particular interest in root bending strength criteria, and surface resistance issues such as pitting and scuffing.

Chapter 4: It gives a summary of the design and show the modifications made to optimise the solution. These modifications are related to the refinement of the calculation with a load spectrum, modification of the tooth profile, assembly of the shafts, analysis of fatigue behavior, detailed and customized choice of rolling bearings, application of spline joints, brief description of housing concerns and presentation of other complementary mechanical components.

Part III is reserved for lubrication and contains Chapter 5.

Capter 5: Chapter 5 is the thesis part exclusively dedicated to the energy dissipation system, selecting the lubricant and its lubrication method. Important considerations are also made regarding the selection of pipe fittings and the definition of minimum flanging lengths and minimum bending radius.

The **Part IV** is dedicated to the conclusion of the work, containing the last chapter, annexes and bibliographical references. The last chapter includes a summary of the work developed and the main conclusions. Some suggestions for future work are also given.

Part I

Chapter 2

State of art

2.1 Introduction

2.2 Brief overview of wind turbine conversion systems

Prior to deciding the prime solution for wind turbine planetary gearbox, it is prudent to have a look at the wind energy conversion systems. Although many people are familiar with the term *energy*, words like *work*, *power* and *fuel* are often used interchangeably with it. Energy is the foundation of human life. In the past, men used muscle power, then fire and animal power. Later, men learnt to harness energy, convert it to useful form and put it to various use.

The wind energy is one of the available forms of energy. It is a free, clean and inexhaustible type of solar powered energy, that is to say, the irregular heating in the atmosphere from the sun, the non-uniformity forms the earth's surface, and rotation of the earth originate a wind flow that provides motion energy to the blades of wind turbine, enabling to generate electricity.

2.2.1 Global wind power

There is hardly any activity that is independent of energy. Its great value was manifested in pursuing new and better sources of energy, such as wind flows. For this reason, it is important to clarify the global distribution of its production and consumption, highlighting the ruling areas of the globe.

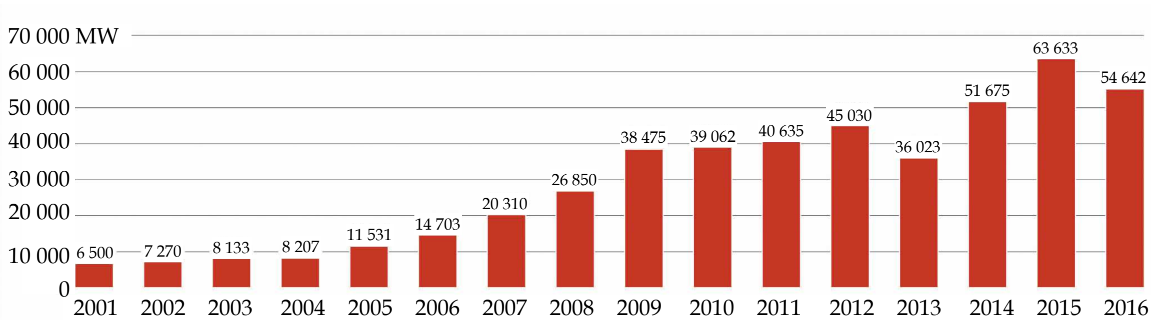


Figure 2.1: Profile evolution of cumulative installed capacity worldwide of 2001–2016, [2].

Taking into consideration the profile evolution of cumulative installed capacity worldwide of 2016, shown in figure 2.1, we can conclude that installed capacity of global wind power has increased exponentially from approximately 6 500 MW in 2001 to 38 475 MW by 2009, remaining stable through the following years. Despite of decrease compared to 2015, in 2016 the wind industry has achieved a trend estimated at approximately 55 GW. There are now 486 749 MW of installed wind power capacity in the world. Figure 2.2 shows the installed capacity by region of 2016 in the world. Unlike what happened in a not so distant past, Asia has exceeded Europe as the largest wind power producer as a region. Its installed capacity has been more than tripled between 2008 and 2016, like predict in [3] the real challenge is to increase the share of wind power in relation to total power generation and Asia leads, once again, with 50.7 % of wind power penetration.Taking a more detailed look at EU market shares for new wind energy capacity installed (fig. 2.3), Germany, France and Netherland lead with over 43.6 %, 12.5 % and 7.1 %.

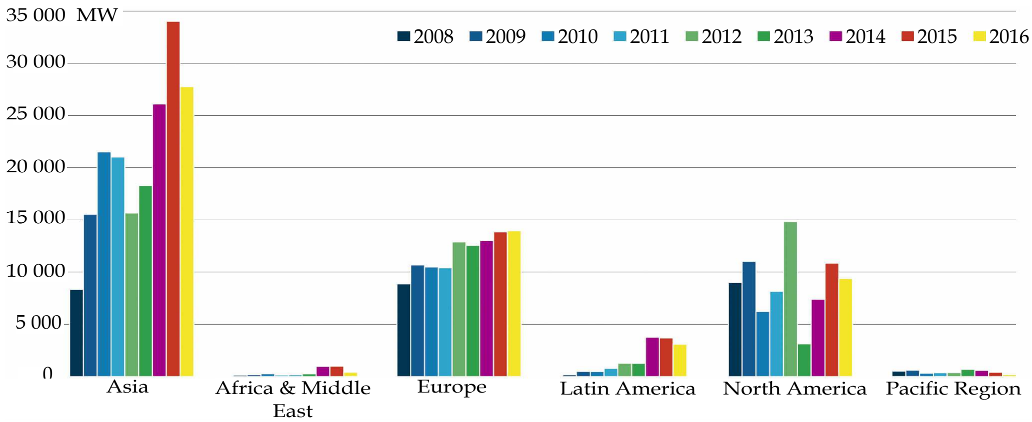


Figure 2.2: Annual Installed Capacity by Region 2008-2016. Total new wind energy capacity globally installed of 54 642 MW. For a more detailed information should be consulted the data source, [2].

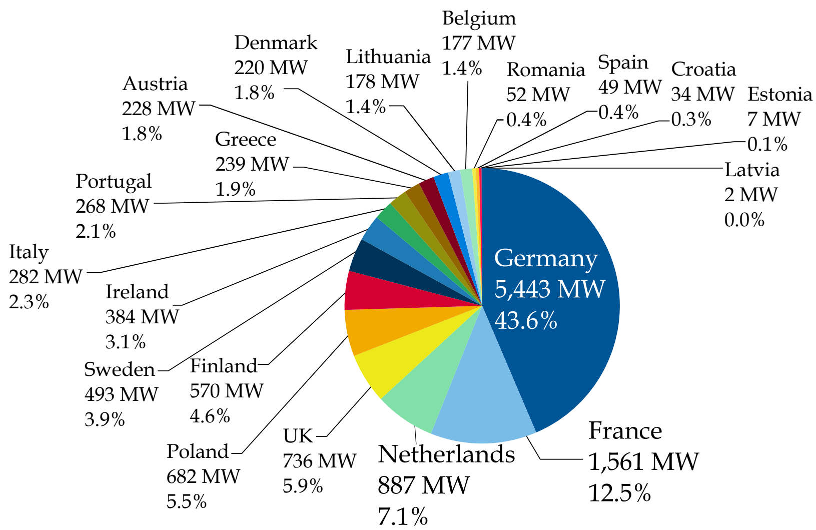


Figure 2.3: EU market shares for new wind energy capacity installed during 2016. Total of 12 490 MW, [1].

The figure 2.4 consolidate the supremacy of wind energy installation programs in support of decommissioning capacity of less environmentally friendly forms of energy, such as coal and oil fuel, or even instable and highly combustible like natural gas.

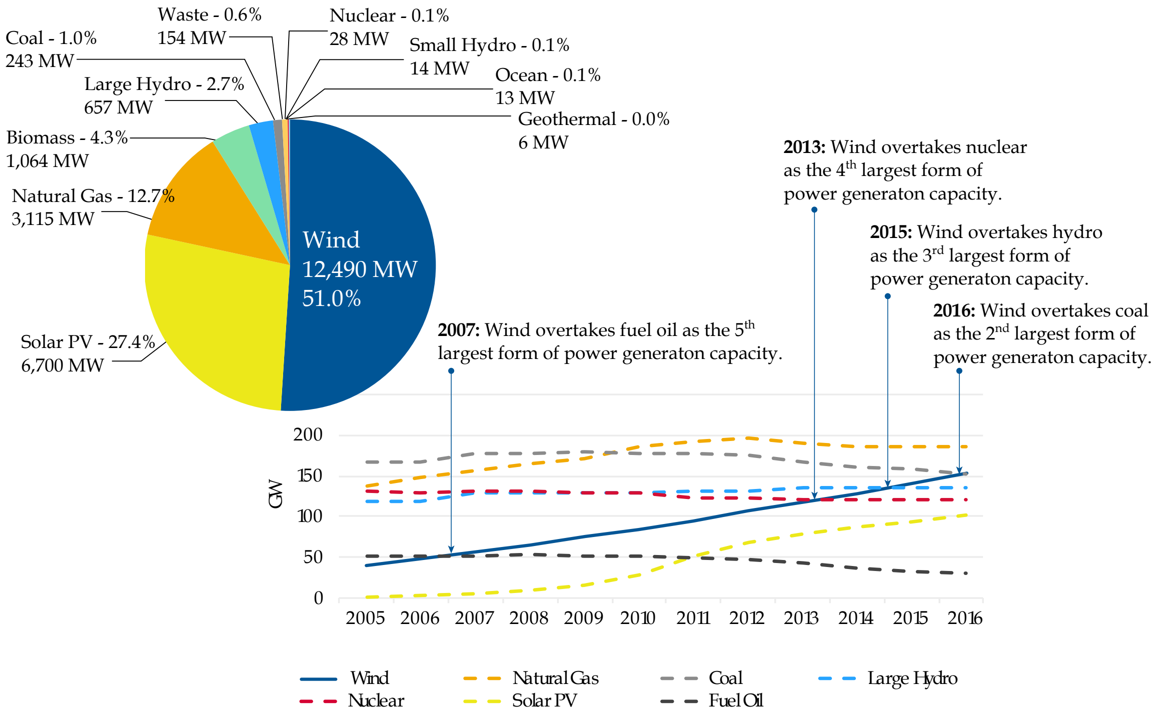


Figure 2.4: At the left, in the circular chart, share of new installed capacity during 2016 (total 24 484 MW). At the right, the cumulative power capacity in the European Union 2005-2016, [1].

2.2.2 Development of wind turbine power

Despite being just a fraction of all electricity globally generated, there had been significant technological and constructive variations to the wind energy industry, in order to satisfy our society energy needs. Over the last decades, installed wind power capacity has been progressively growing, ranging from a few kilowatts for residential or commercial use to a several megawatts in large wind farms.

On one hand, for remote areas, where access to the power grid is difficult or costly, small wind power units can be used in combination with others energy sources such as photovoltaic power and diesel generators to form a standalone generation system.

On the other hand, the size of large wind turbines has progressively increased, giving the lead of wind Energy Company's race to the 8 MW Vestas V164 turbine . The evolution in turbine size can be clearly appreciated in figure 2.5. Notice that, rotor diameter and power rating of SeaTitan WT is somewhat bigger than the aforementioned one, however it is only a prototype and so it is not yet in application.

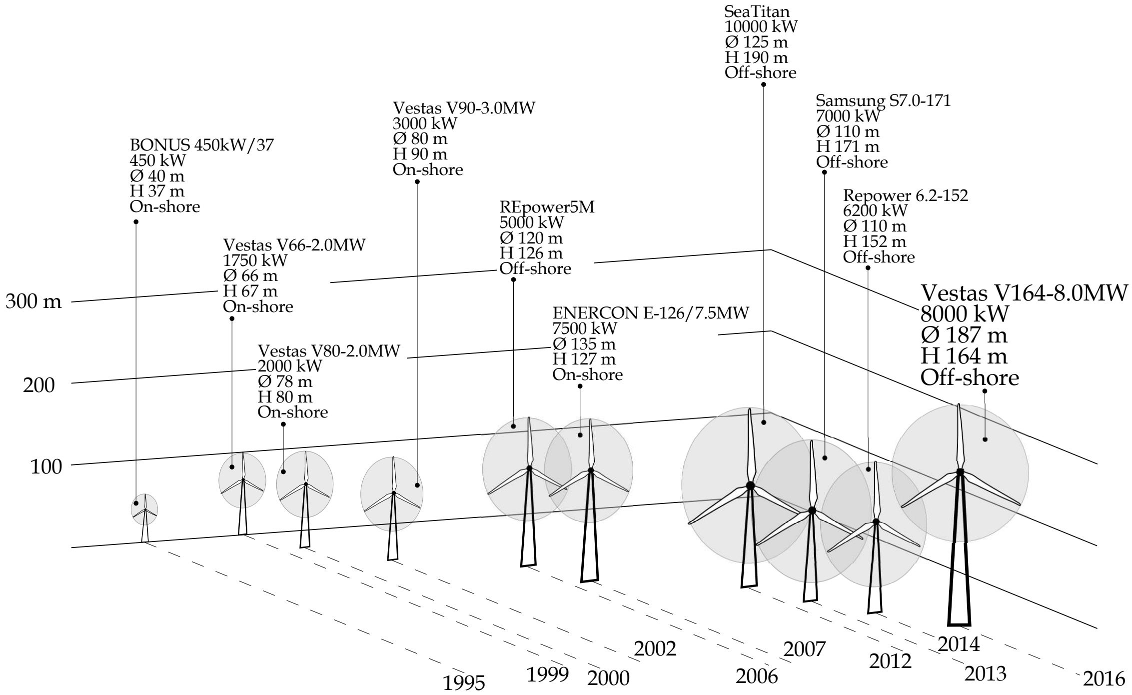


Figure 2.5: Evolution of wind turbine size (Ø: rotor diameter; H: tower height). All data can be founded in different articles, manufactures reports and datasheets, [3–11].

2.2.3 Costs of wind turbine conversion systems

Energy request is expected to increase widely during the following decades. Naturally this is creating panic that our energy assets are beginning to run out and a worldwide economy worry stands up. As a result of that, we must take account of the cost of energy.

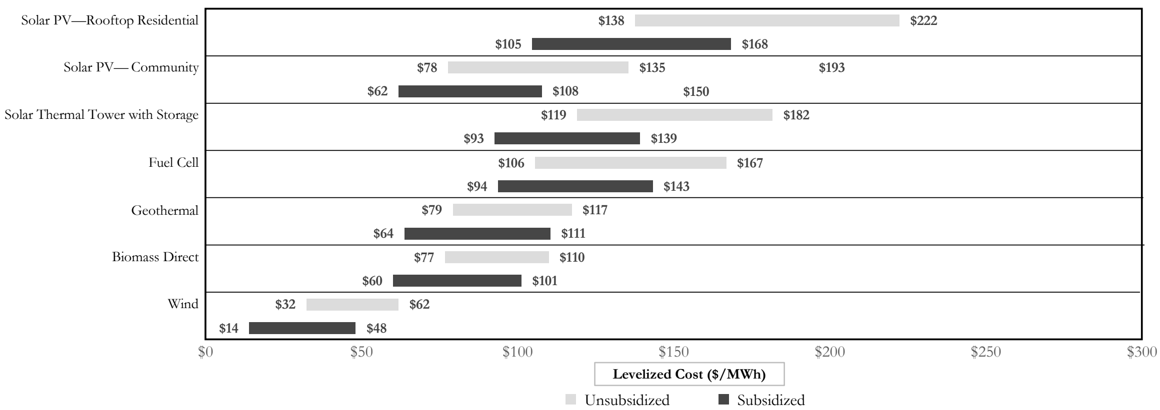


Figure 2.6: Leveled cost of energy in United States. U.S. federal tax subsidies remain an important component of the economics of Alternative Energy generation technologies (and government incentives are, generally, currently important in all regions), [12].

When the first utility-scale turbines were installed in the early 1980s, wind-generated electricity cost was $ 0.3 per kWh, [3]. Today, as we can see in the figure 2.6, wind power plants can generate electricity for $ 0.032 to $ 0.062 per kWh, [12]. Compared with other energy resources, alternative or conventional ones, wind energy is one of the most economically viable renewable energy resources, as illustrated in figure 2.6. Variations in power rating, operating condition, location and technology used can materially affect the level of the cost of energy production.

Some of the key parameters governing wind power economics are the: design decisions and project financing, operating and maintenance costs, and economic lifetime. Considering that around 7.2 % of the wind turbine cost is directly related to the gearbox, the more manufacturing gearbox we have, the more suitable wind turbine idem. Table 2.1 gives the cost analysis of a typical wind turbine.

Table 2.1: Cost breakdown of a typical wind turbine, [13].

Categories	Component	Share o total cost %
Turbine *	Tower	16.4
	Rotor blades	14.8
	Gearbox	7.2
	Power Converter	3.5
	Transformer	2.6
	Generator	2.9
	Other	16.6
Foundation		16
Planning and Miscellaneous		11
Grid connection		9

* the turbine by itself account for around 64 percent of the overall cost of the wind energy conversion system.

2.3 Wind turbine technology

2.3.1 Configurations and main components of a wind turbine

A burst of new wind turbine design and configuration has been brought about by worldwide interest in renewable energy option, and modern wind generators fall straight into two fundamental groups: the horizontal-axis wind turbines (HAWT), which the spin axis of the wind turbine is parallel with the ground, and vertical-axis wind turbine (VAWT), which the orientation of the spin axis is perpendicular to the ground [3], figure 2.7.

HAWT dominate the majority of the wind industry because of their higher wind energy conversion efficiency due to the blade design and access to stronger wind. However, VAWT have their place, in small wind and residential wind applications. It has the advantage of lower installation cost and easier maintenance, due to the ground-level gearbox and generator installation.

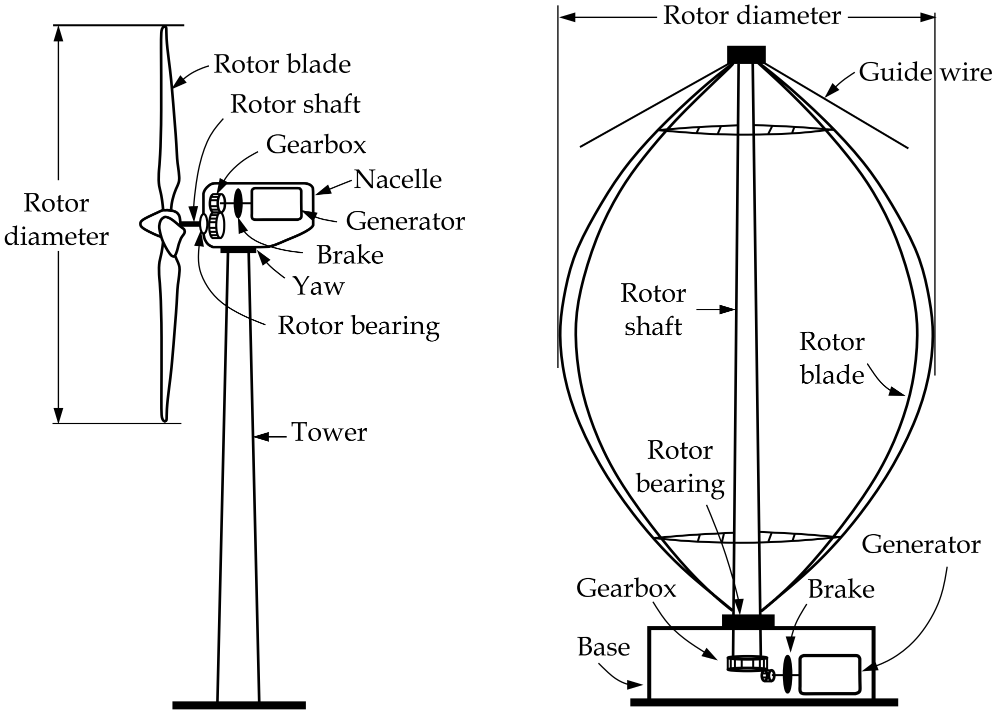


Figure 2.7: *Horizontal* and *vertical* axis wind turbines, [3].

According to [14], horizontal turbine components include:

- blade or rotor, which converts the energy in the wind to rotational shaft energy;
- a drive train, usually including a gearbox and a generator;
- a tower that supports the rotor and drive train; and
- Other equipment, including controls, electrical cables, ground support equipment, and interconnection equipment.

Blades and Rotor hub The area of the wind generator that accumulates energy of the wind is known as the blades. The wind turns the blades, converting the linear motion of the air into circular motion. The Rotor hub is responsible to connect the blades to the drive train, powering a power generator that supplies a current.

Drive train The generator converts the particular rotation of wind turbines blades directly into electricity. It generally demands rpm's of just 1500 to at least 1800, which is far higher than the amount of revolutions per minute of wind turbine blades range. Consequently, most WTs requires a gearbox. It turns the high torque, low speed of the wind into a low torque, high speed of the generator.

Tower and Nacelle The tower elevates the nacelle to provide sufficient space for the blades rotation and to reach better wind conditions. The nacelle supports the rotor hub and houses the gearbox, generator, and some others equipment like previously stated.

2.3.2 Power controls

Wind turbines can also be classified into fixed-speed and variable-speed turbines, achieving maximum conversion efficiency only at a given wind speed and over a wide range of wind speeds, respectively. In fixed-speed wind turbines, the constant speed that it rotates is determined by the transmission ratio, the grid frequency, and the number of the poles of the generator, [3].

Turbine blades are aerodynamically optimized to capture the maximum power from the wind, the most commonly used methods being pitch and stall controls.

For a robust and simple solution, without mechanical actuators, sensors or controllers, the control method tipically used is the passive stall control. Regarding the stall control, the blades of the turbine are designed, such that, when the wind speed exceeds the rated wind speed of about 15 m/s, air turbulence is generated on the blade surface that is not facing the wind, resulting in a reduction in the captured power and preventing turbine damage.

Conversely, pitch control, which is used in wide-scale wind turbine, turns the blades in their longitudinal axis, changing the pitch angle through a hydraulic or electromechanical device located in the rotor hub attached to a gear system ate the base of each blade.

Both are used to reduce the captured power at high wind speeds, which prevents turbine damage under wind gusts effects. Note however, pitch control shows versatility to constrain captured power over a wide range of linear motion of the air and, therefore, ensured a more profitable WECS. In cases in which the wind speed is higher than the limit of about 25 m/s, the blade are pitched completely[1] out of the wind, and thus no power is captured, [3].

2.4 Gearbox in a wind turbine

As concluded in section 2.2.3, the gearbox is a costly part of an wind turbine, and thus *direct drive*[2] generators that operate at lower rotational speeds are being explored. In that case, the wind turbine must be able to continuously adjust its rotational speed in order to keep the tip speed ratio at an optimal value to achieve the maximum power conversion efficiency at different wind speeds. To make the turbine speed adjustable, the wind turbine generator is normally connected to the utility grid through a power convert system. So, in whatever way, this is a dimension, technologic and economical demanding task and that is why the ***gearbox-generator*** system continues to be the most usually.

[1] it is also known as fully pitched or feathered.

[2] in these machines, the generator rotor turns at the same speed as the turbine rotor, dispensing a gearbox.

2.4.1 Description

Due to the loading and environmental conditions in which the gearbox must operate, its design is a huge challenging job. In fact, for example, it may be exposed to an ambient temperature range of -29 °C to +50 °C [3], [15].

To address the operating constraints, the design of the gearbox shall be made in accordance with IEC 61400 Wind turbine standard. For the current calculation of fatigue resistance, this standard suggests to have a look at guide values of gear and bearing standards by ISO/TR. For instance, the ISO 6336 gear standard provides an established method for calculating resistance to subsurface contact failure and for tooth root breakage but these are neither the only nor the most usual failure modes in WT gearboxes. In fact, for this kind of application, the gearbox collapse depends for the most part on: manufacturing errors, such as grind temper or material inclusions; surface related problems; and, problems from small vibratory motions. Along with poor lubrication, such issues can cause surfaces overheating and its subsequent adhesive wear. This meet scuffing, micropitting, and fretting problems, resulting in the detachment and transfer of particles from one or both of the meshing teeth, [16]. So, information on some of these specific troubles can be found in the fourth part of IEC 61400 standard. Many wind-turbine gearboxes have also suffered from fundamental design issues such as ineffective interference fits that result in unintended motion and wear, ineffectiveness of internal lubrication paths and problems with sealing.

Regarding the operating revolutions, the rotor of a large three-blades wind turbine usually operates in a speed range from 6-20 rpm, [3], what is much slower than a standard wind generator with a rated speed about 1500 or 1000 rpm and 1800 or 1200 rpm, with a 4 or 6 poles for a 50 Hz and 60 Hz generator frequency, respectively. So, a gearbox is needed to increase rotational speed from a low-speed rotor to a higher speed electrical generator. This is because of the power grid frequency, typically, 50 Hz in Europe and 60 Hz in the United States.

For a given generator frequency in Hz, f, and number of poles, np, the generator shaft speed in rpm, n_{gen}, can be determinate by:

$$n_{gen} = \frac{120f}{np}. \tag{2.1}$$

The gearbox conversion ratio[4], i, is designed to match the high speed generator with the low speed turbine blades, and it is given by them ratio,

$$i = \frac{n_{gen}}{n_{rotor}}. \tag{2.2}$$

So, considering the more typical cases of a 4 or 6 pole generator, the transmission ratio is described by the several curves presented in figure 2.8.

[3] or -20.2 °F (244.15 K) to 122 °F (323.15 K). The ambient temperature is defined as the dry bulb air temperature within the nacelle in the immediate vicinity of the gearbox

[4] it is also known as *transmission ratio*.

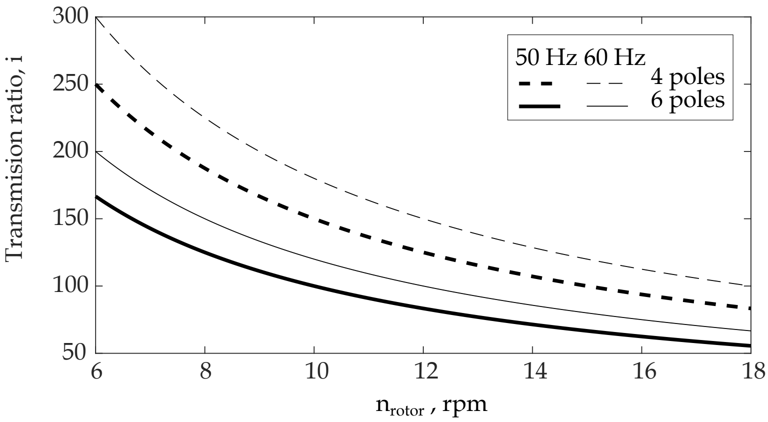


Figure 2.8: Transmission ratio i versus the rated turbine speed n_{rotor}, [3].

Among the different configuration, like spur or helical gears and parallel or planetary stages, two or more gear types and gear stages may be combined to achieve the high conversion ratio needed to couple the turbine rotor and the generator.

Based on a rotor speed of around 15 rpm, we can claim that, for a 1500 rpm output for the generator, a 100:1 transmission ratio is considered and it would play a pivotal rule in the stages layout selection. Most installations have one generator, i.e. the gearbox has exactly one input and one output. Although the chosen design is up to manufactures, it is common to find ***one planetary-two helical stages*** transmission for lower power gearboxes. In multimegawatt WT, gearbox are more likely to have a *multi-stage planetary* arrangement, than the few megawatts ones. Epicyclical gear stages provide high load capacity and compactness to gear drives offering a multitude of gearing options and a large change in RPM within small volume. The limitation of planetary gearbox is the need for highly complex design and the general inaccessibility of important parts and high loads on the shaft bearings. Cooling of the gearbox is done by the gearbox oil, [17].

2.4.2 Concepts

Considering the gearbox mission and according to their kinematics type, wind turbine gearboxes can be classified as follows, [18]:

- **standard gearboxes** that constantly transform input torque and speed values into those at the output;
- ***torque-limiting*** **gearboxes** can limit the output torque and, as a consequence, limiting the input torque. The speed is thereby not controlled.
- **CVTs gearbox** that permit the control of the gearbox output speed and torque[5].

As described above, i.e. bearing in mind that *one input-one output* configuration is commonplace, most of the WT gearbox assume one of the following described configurations.

[5]within a determined range.

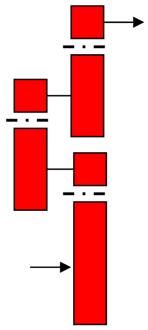


Figure 2.9: Three spur wheel stages arrangement, [18].

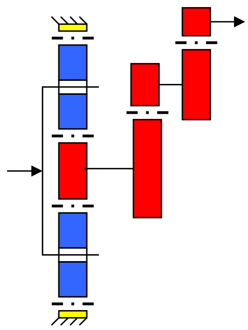


Figure 2.10: One planetary-Two parallel stages gearbox, [18].

The *three spur wheel stages* gearbox (see fig. 2.9) is a typical concept for the lower capacity generator. As the name suggests, the single transmission ratio of each stage is given by the ratio between its input speed and its output speed, or directly by the ratio between the number of gear teeth like shown by the Equation (2.3). The *three spur wheel stages* gearbox , like all other examples, normally works with helical gears for lower noise.

$$u_{\text{three spur wheel}} = \frac{n_{out}}{n_{in}} = \frac{\frac{v_{out}}{r_{\text{out}}}}{\frac{v_{\text{in}}}{r_{\text{in}}}} = \frac{r_{out}}{r_{in}} = \frac{Z_{out}}{Z_{in}} \tag{2.3}$$

In the *one planetary-two parallel stages* arrangement (fig. 2.10), the sun gear[6] is engaged with a portion of the planet gear[7] that is in mesh with the stationary ring gear[8], commonly ring gear is part of casing. In this type of meshing it is require a offset in the layout of the gear stages.

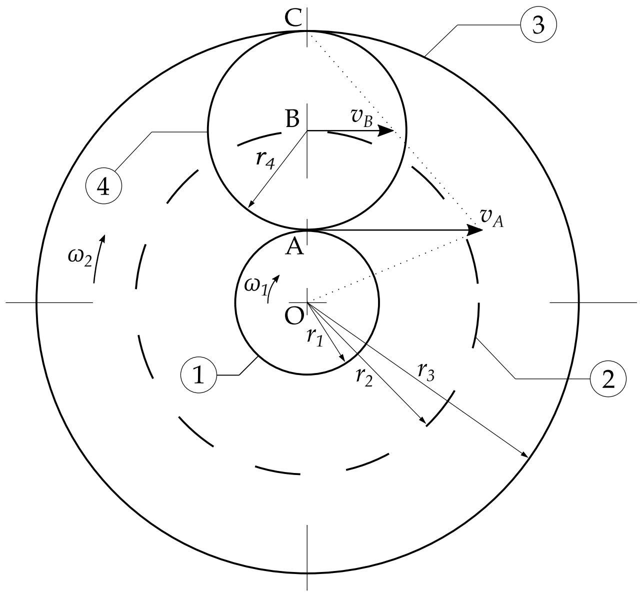


Figure 2.11: Kinematic diagram of a fixed ring epicyclic gear drive.

[6]often named just as *sun* during this work.
[7]often named just as *planet* during this work.
[8]due to its internal meshing, often named as *internal gear*. It can also just be called *ring* in this thesis

Taking into account the stationarity of the gear ring **3** and the kinematic diagram of the figure 2.11, based on the kinematic diagram of the Figure 2.11 the circumferential speed at the point A can be established by,

$$v_A = r_2\omega_2, \tag{2.4}$$

which is related to the linear speed at point B by the following relation,

$$v_B = 2v_A. \tag{2.5}$$

It is worth noting that the distance from point C to point B is twice the distance between points C and A. Since there is a linear relationship with the circumferential speed of a point and it's the straight distance from the instantaneous centre of rotation, the Equation (2.5) is true.

Analysing the Figure 2.11 we can conclude, too, that the linear speed of the point B is also given by,

$$v_B = r_1\omega_1. \tag{2.6}$$

With the definition of the gear pitch radius,

$$r = \frac{Zm}{2} \tag{2.7}$$

where Z and m are its number of teeth and nominal module, respectively, we can obtain the following expression by matching the Equations (2.5) and (2.6),

$$\begin{aligned}
2v_A &= r_1\omega_1 \\
2r_2\omega_2 &= r_1\omega_1 \\
2\left(r_1 + r_4\right)\omega_2 &= r_1\omega_1 \\
2\left(r_1 + r_4\right)n_2 &= r_1 n_1 \\
2\left(\frac{Z_1 m}{2} + \frac{Z_4 m}{2}\right)n_2 &= \frac{Z_1 m}{2} n_1 \\
2\left(Z_1 + Z_4\right)n_2 &= Z_1 n_1 \\
\left(2Z_1 + 2Z_4\right)n_2 &= Z_1 n_1
\end{aligned} \tag{2.8}$$

Substituting the relation $Z_3 = Z_1 + 2Z_4$ in equation (2.8), it can be rewritten,

$$\begin{aligned}
\left(2Z_1 + Z_3 - Z_1\right)n_2 &= Z_1 n_1 \\
\left(Z_1 + Z_3\right)n_2 &= Z_1 n_1
\end{aligned} \tag{2.9}$$

Finally, according to the previous Equation (2.9), the transmission ratio for the planetary stage of the mentioned gearbox concept is given by,

$$i_{\text{one planetary-two parallel}} = \frac{n_{out}}{n_{in}} = 1 + \frac{Z_3}{Z_1} \tag{2.10}$$

and its practical range varies from 3:1 to 9:1.

Table 2.2 shows how many planets can be engaged for the various gear ratio $\frac{Z_{ring}}{Z_{sun}}$. Even if the number of planets is usually mentioned to as a purpose of the transmission ratio, it is very common to use only three planet gears. This solution allows to take advantage of the torque splits, forces and moments stability, which characterize the epicyclical gear, without making the geometric tolerances too tight.

Gear ratio $\frac{Z_{ring}}{Z_{sun}}$	**Possible number of planets**
12.0	3
5.2	4
3.4	5
2.7	6
2.2	7
2.0	8

Table 2.2: Possible number of planets in relation to the gear ratio Z_{ring}/Z_{sun}, [19].

Let's look at torque splits in terms of fixed support and floating support of the members. With permanent support, all members are supported in bearings. The centres of the sun, ring, and planet carrier[9] will not be coincident due to manufacturing tolerances. Because of this, less planets are simultaneously in mesh, compromising the equitable share load by all planets. With floating support, the members are allowed a small amount of radial freedom or float, which lets the sun, ring, and carrier to seek a position where their centres are coincident. This float could be as little as about 25–50 μm[10]. With floating support three planets will always be in mesh, resulting in a higher effective number of planets sharing the load, [20]. Irregular angular positioning of the planet gears may result in an imbalance in the planetary stage. This must be avoided by carrier design, dimensioning and assembly.

It should be noted that the more planets we have, the greater dimensional deviations between the different meshing members idem. It would require extremely high manufacturing quality to ensure exactly the same dimensions between them.

Figure 2.12 shows the *two planetary-one parallel gear* stage arrangement, with the sun gears of both stages housed in different shafts and the ring gears of both stages connected to the housing gearbox. The shafts supporting the first and the second stages' planet carriers are typically connected to the gearbox housing by rolling bearings, allowing its rotary motion. Here, the planet carrier of the first stage is connected to the input shaft and simultaneously engaged with its ring and sun gear. Ideally, the first-stage sun gear is housed in the same shaft of the second stage's planet carrier, being excited of the equal rotating speed. The same occurs in the second gear stage, transmitting the rated power from the planet carrier to the gear of the last parallel transmission stage of the gearbox. Application of the *two planetary-one parallel gear stage* allows very high transmission ratio values to be achieved. In this planetary arrangements, the transmission ratio usually does not exceed 100:1, [20].

[9] also known only as *carrier*.

[10] or 0.001–0.002 inches.

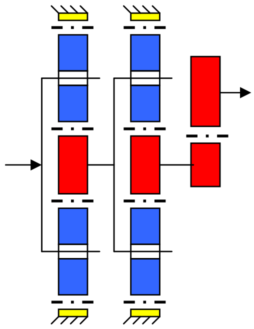


Figure 2.12: Two planetary-One parallel gear stage gearbox, [18].

Figure 2.13 shows *compound* planetary gearbox with the planet carrier of the first stage and the ring gear of the second stage connected to the same driven “shaft”. Contrary to the above example in which 100 % of power flows through all stages, in the *compound planetary* gearboxes, we have internal load splitting. It enables to increase the transmission ratio, because of the diminishing of the load meshing per gear stage due to the load shared between planetary stages.

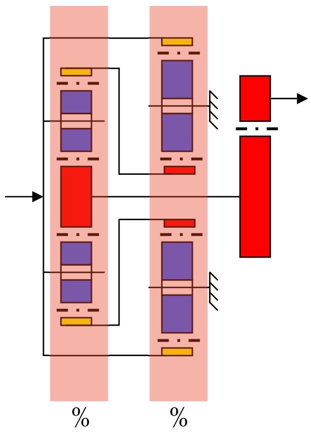


Figure 2.13: Compound planetary gearbox, [18].

Despite the simplest lubrication motivated by the fixed planet and the higher transmission ratio above mentioned, it is an exceedingly complicated and demanding constructive solution, because of difficulty associated to the drive system of the ring gear and because of the axial loads that this involves on planet carrier.

For the last, figure 2.14 displays a *differential planetary* gears arrangement in which, in the first stage, ring gear is fixed and carrier is driving [11], and which, in the second stage, ring gear are driving the sun through the planets that are fixed[12]. In this kind of gear arrangement, shown in figure 2.14, ring and carrier are driven by previous stages in a multiple performance. In *differential planetary* arrangements with compound planet gears, operating pressure angles in the planet/stationary ring gear mesh and in the planet the planet/rotating ring gear mesh can be selected independently. This allows for balancing specific sliding velocities in these meshes to maximize gear efficiency, [20].

[11]also called *epicyclic set*.
[12]also known as *stationary set*.

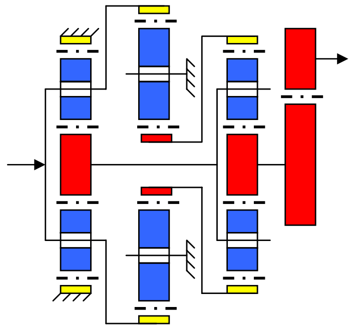


Figure 2.14: Differential planetary gearbox, [18].

In the following figure 2.15 some others gearbox concepts are illustrated.

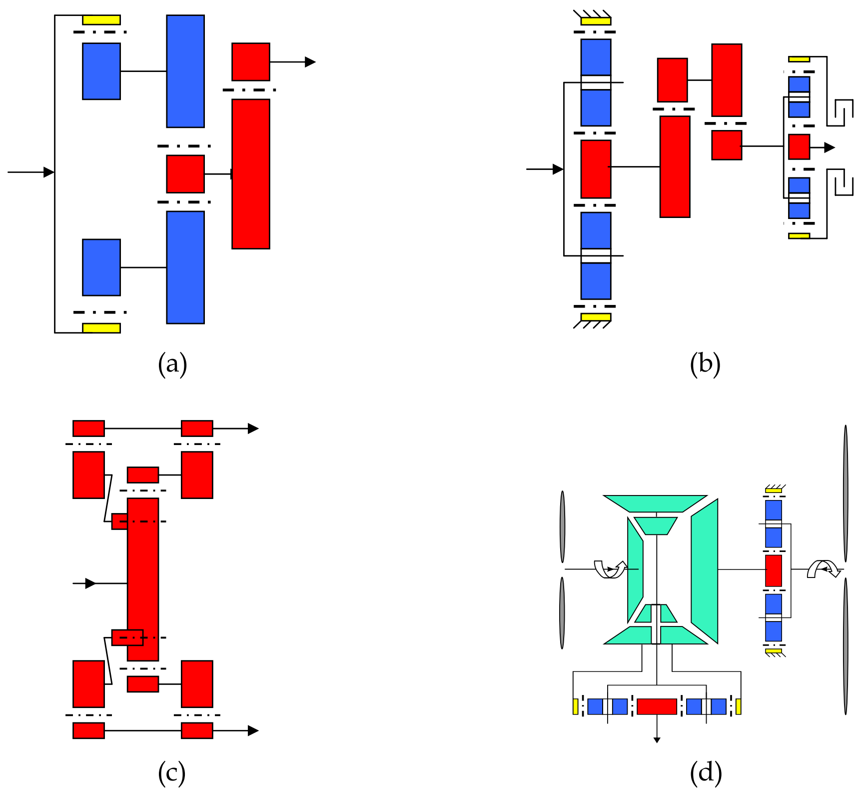


Figure 2.15: Several gearbox concepts: (a)Planetary gear stage, fixed Planet: Renk Aerogear; (b) Hydraulic torque limiting: Henderson Gearbox; (c) Power Split, several Generators: Clipper ; (d) Two inputs: Luv and Lee Rotor, Kowintec, [18].

2.4.3 Market search

Parameters like *wind turbine operating power* and *rotation speed of the rotor (or blades)* are influencing parameters in the gearbox conception. A conscious choice must be made according to the needs of today's energy suppliers. The aim of the market share analysis is to obtain the trend lines of the most important parameters between the main company that originally built and work with wind turbine technology.

In this context, it is once again essential to design an applicable transmission group; thus, the market research proves to be relevant, in a way it allows the mapping out of the dimensions and mass values, as well as, its suspension mechanisms.

As reported by the statistics portal Statista, Vestas reclaimed the top spot in the annual ranking of wind turbine manufactures, with a 16 percent of the world's market share for wind turbines. General Electric paired with Goldwind placed second with 12 percent, followed by Gamesa (8 %), Enercon (7 %), Siemens (6 %), and like many others.

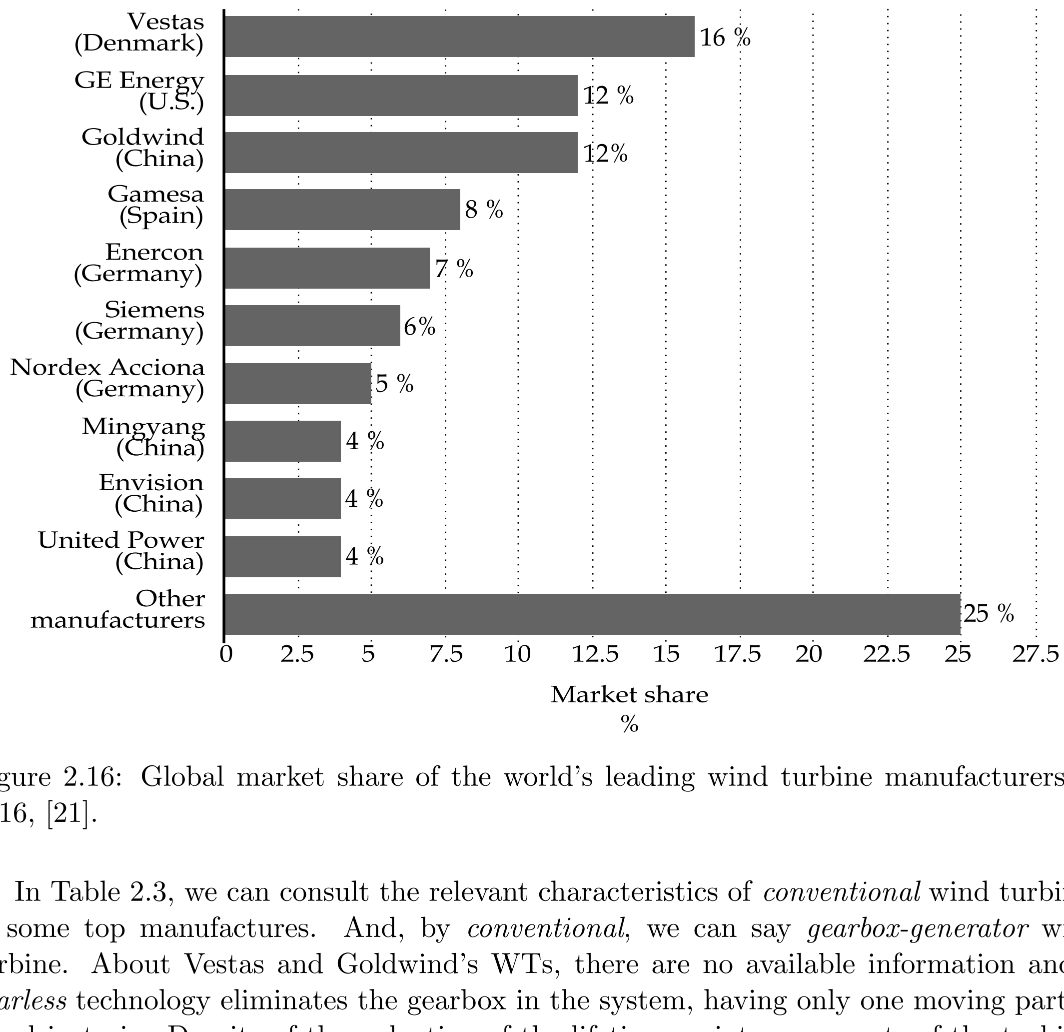


Figure 2.16: Global market share of the world's leading wind turbine manufacturers in 2016, [21].

In Table 2.3, we can consult the relevant characteristics of *conventional* wind turbines of some top manufactures. And, by *conventional*, we can say *gearbox-generator* wind turbine. About Vestas and Goldwind's WTs, there are no available information and a *gearless* technology eliminates the gearbox in the system, having only one moving part in the drivetrain. Despite of the reduction of the lifetime maintenance costs of the turbine, its manufacturing process and technologic development is extremely expensive, compared to a conventional one.

Table 2.3: Top manufactures wind turbine characteristics, [22; 23].

Parameter	Unit	Value		
Manufacturer		Gamesa		Siemens Wind Power
Model		G87 - 2.0 MW	G97 - 2.0 MW	SWT - 2.3 - 101
Generator power	kW	2000	2000	2300
Rotor diameter	mm	87	97	101
Operating interval	rpm	9.0–19.0	9.6–17.8	6–16
Gearbox configuration		1 planetary/2 parallel stages		3 planetary stages
Gearbox ratio		100.5:1 (50 Hz) 120.5:1 (60 Hz)	106.8:1 (50 Hz) 127.1:1 (60 Hz)	91:1 *

* grid frequency not specified

Discretizing wind turbine into its different components and, disaggregating and emphasizing the gearbox, according to [24], the most representative sellers in the market are:

- China High Speed Transmission;
- Gamesa Energy Transmission;
- Moventas;
- Winergy;
- ZF Friedrichshafen.

Having an overview of the detailed information in the tables above, for a input power of about 2 MW, the predominant transmission group configuration seen is ***one planetary/two parallel stages***, Tables 2.4 and 2.5. We also found that, for this power range, the order of magnitude of the wind turbine gearboxes mass is around tens of tons with an overall length of approximately 2500 mm; the operating speed is between 6–19 rpm; and, the transmission ratio is mainly contained in the range [91:1–120:1].

Table 2.4: Detailed ZF Friedrichshafen gearbox information and parameters, [25].

Parameter	Unit	Value		
Manufacturer		ZF Friedrichshafen		
Topology's model		3 point suspension	4 point suspension	
Rated power	kW	2100	2000	2050
Gearbox configuration		1 planetary/2 parallel stages		
Approx. weigh	kg	17 000	15 000	19 000
Overall length	mm	2451	2210	2677

Table 2.5: Detailed GAMESA gearbox information and parameters, [23].

Parameter	Unit	Value	
Manufacturer		GAMESA	
Model		2.0i101	2.0i107
Rated power	**kW**	2 000	
Input speed	**rpm**	16.63	15.73
Gearbox ratio		101.0:1	106.8:1
Approx. weigh |*	**kg**	14 650	
Overall dimension |**	**mm**	2 339/1 714/2 210	
Oil grade		ISO VG 320	
Oil fill quantity	**l**	315	

* dry mass
** width/height/length

By means of introduction, it should be noted that in GAMESA gearboxes, seen in Table 2.5, 315 liters of lubricating oil ISO VG 320 is required. This issue is still ongoing and it will be covered in section 3.1.11, about the lubrication system of this wind turbine gearbox.

Part II

Chapter 3

Gearbox Design and Dimensioning

This section is especially set up for the purpose of (i) defining the structure of wind turbine; (ii) clarifying and explaining the choices of the layout and design of its transmission component; and, (iii) dimensioning the mechanical elements. The dimensioning of the mechanical elements is based mainly on KISSsoft® calculations.

3.1 Transmission

Before we can discuss the different type of gear, it shall be defined what *gear* means. In more technically correct terms as possible, a gear is a toothed wheel that is usually, but not necessarily, round, [26]. Used with the purpose of transmitting motion and/or power from one shaft to another, there are multiple varieties of profiles that its teeth may have. So, it's prior to define that a *involute-based form* was considered, at least for parallel axis gears, the most common tooth form, [26]. *Involute-based form* performs simple manufacturing, low flexibility to small changes in center distance, strength and durability, that other "involute brothers" don't present. Given its geometrical and kinematic complexity, it is essential to define notation symbols for the data processing relating to its resistance and meshing calculation. By meshing is meant a set of two coupled toothed wheel. So, the insulated element is called a *gear* and the one with the lowest number of teeth is often known as *pinion*, [27].

The nomenclature used in this paper is suggested by International Standards Organization—ISO 701:1998.

3.1.1 Number of transmission stages

The definition of the kinematic chain of the gearbox was the first step of the design in this work, and it was fundamental in all subsequent decisions taken. The first decision to be made was based on the number of stages required to achieve the desired speed increase. As suggested in the previous chapter, given the high order of magnitude of the turbine input power, it would be unfeasible a solution of only one transmission stage. Considering that four stages of multiplication speed would force more mechanical elements that introduce power losses in the system, we have the solutions of two and three transmission stages.

According to the graph of the figure 2.8 and considering a nominal rotor speed of the around 15 rpm, it can be seen that the range of the conversion speed ratio for these gearboxes is roughly 60:1 to 150:1. The choice of a *two stages* solution could translate into a transmission ratio for the first stage of at least 8:1. This would result in a very severe multiplication of the speed, and hence a very severe torque reduction, in each of the stages. This could cause increased power losses due to uncontrolled sliding effects, consequence of the high gear modules and the high rolling bearing size required.

That said, in order to ensure a more energy-dimensionally efficient solution, the *three stage* conversion solution would be chosen.

3.1.2 Shaft arrangement

3.1.2.1 1st Stage

The gears can be classified based on the arrangement of the axes of the gear pair, which transmit the rotational movement. In this work, parallel axis gears are fully used, and in the first stage of speed multiplication, a particular case of it is considered—The epicyclic gear drive.

Epicyclic gear are increasingly used. Having undertaked a brief study of these mechanisms, indicating mathematical and graphical methods of calculations in Section 2.4.2.

It will be showed that it should not be used early, too. This because in certain configurations we would risk achieving extremely low meshing efficiency, [28].

In this respect, an epicyclic gear drive is composed of the following main parts:

- Two coaxial shafts with different rotational speeds and on which are mounted the sun gear and the ring gear, respectively;
- A articulated chassis which turns about an axis corresponding to the axis of the previous shafts. It rotates at its own speed and supports the planet gears, providing a connection between the other two coaxial shafts.

3.1.2.2 2nd and 3rd Stages

Having decided the gear arrangement for the first stage, focus can shift to the second and third stages. It has been considered the WT gearbox as a two single gearboxes coupled, one planetary and two parallel shaft stages. This decision was made considering that there was already done a foregoing work about the second part of the gearbox. In other words, a numerically proven comparison of he different parallel shaft arrangement was previously been presented at the master thesis of João Pedro Sousa, [29]. Thus, the starting point for the selection and definition of the gear stages in question shall be the results of that work.

As stated by [29], the parallel stages offer significant flexibility in their configuration and, so, the *single branch* and *double branch* arrangement were studied and compared. The *single branch* is the most convectional arrangement of the transmission, being such a widely applicable as a low cost design and manufacturing.

An alternative is *double branch* arrangement. Through the division of the rated power, this solution presents a robustness far superior them the previous one. That is, with the *double branch* it is allowed to split the power by several paths, where each of them only knows a percentage of the total power of the system. In this way, considering each of the gear paths separately, we are facing a configuration with a appreciably lower magnitude of size, working with lower nominal modules than the *single branch* arrangement. This not only allows a more compact and robust solution as an extremely advantageous solution in terms of maintenance costs. It must be noted that it has the major advantage that in case of failure of the components after the torque splitting, these components can be removed for repair, keeping the turbine partly operational until the repair is completed.

3.1.3 Gear type and Helix angle

It is important to stress that the spur gears are the most common and most used type of gear; however, helical teeth enter the meshing zone progressively and, therefore, have a smoother action than a spur gear teeth and tend to be quieter, which is a major added value according to the problem definition set in Section 1.3. This is because, at helix angle greater than zero, it is possible to get several axial pitches, fig. 3.1, within the facewidth of each helix, allowing higher contact ratio between both meshed gear. It is also known that smaller helical gears have the same load carrying capacity, than a spur gear. For that, helical ger teeth are fully used in this WTG.

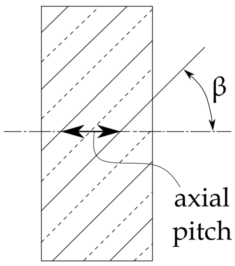


Figure 3.1: Axial pitch of a helical gear.

Having in mind that parallel axis gear is the most efficient type (or form) of gearing and used in this gearbox, *single helical* and *double helical* gear shall be studied and compared.

3.1.3.1 1st Stage

Regarding the first stage, we would do well to start clarifying that planetary stages have simultaneously internal and external meshing; so, we have to be aware that they both must have the same helix angle, but be of different and same hand, respectively.

As for the double helical teeth, there is no far application in the planetary gear type because of the difficulty of construction, [30]. For that, and for a rather extensive amount of time required for few improvement, a *single helical* gear was considered. It should not be forgotten that to provide the helix's improvement in the transmission, the contact ratio provided by the helix angle |[1], ϵ_β, must be at least unity, otherwise for analytical purposes the gear is treated as if were a spur gear.

[1]according to ISO, it can also be called *overlap ratio*; by AGMA, we often found *face overlap*.

As stated in preceding section, a decreasing of the noise level and a increasing of the load capacity is verified with the implementation of the helix angle; however, at angles much above 15° to 20°, the strength of the root teeth is compromised because of the decrease of the transverse tooth thickness, [26]. Having this into account, a helix angle of 15° was chosen.

3.1.3.2 2nd and 3rd Stages

Based on the study comparison introduced by [29], a *double helical* gear was chosen for the second part of the gearbox. Although the slightly dimensional disadvantage identified, a important profit was registered about the power losses in the rolling bearings.

That is, the helical gears produce an end thrust along the axes direction, forcing the application of the tapered roller bearing. In fact, this kind of rolling bearing have high values of power losses, at a high rotational speed. Since the second and third stages are those with the highest gearbox speed, it is interesting to avoid their use. Then, with the double helical gear chose, we obtain the noise benefits of the single helical gears without the disadvantage of thrust loading and, because of that, a higher helices angle could be considered.

Despite the smoother operation, higher helix angles provide tooth strength diminishing. This being the case, the selection of helix angle must be carefully evaluated to ensure that it is strong enough to take the loads in question.

When the gears are made double helical, there is no thrust load because the thrust of one helix is opposed by an equal and opposite thrust from the other helix, [26]. Attention must be paid to the balance of the forces between each of the double teeth. If the stiffness of the double helical gear with its own shaft is too high, the division of load in a double helical gear may be impaired, due to the manufacturing errors and disparities. And so, a *float* configuration must be applied. Il will require a higher helix angle, in order to provide a reduction of resistance to axial float resulting from friction in coupling devices (like shaft). In order to achieve this decreasing of frictional resistance without compromising the root teeth resistance, helix angle of 25 ° was chosen.

3.1.4 Module and number of teeth

The choices of the module values and the number of teeth associated with each stage were made in the sense of minimizing the size of its gears, as well as increasing its efficiency. In addition, the ratio between the number of teeth in each pair, i.e. the gear ratio of each stage should be a rational number, namely a periodic fractional number, so as to minimize repeated contact between the same pair of teeth, which can suppress the number of working teeth, promoting the propagation of wear failures, or even, the root breakage of teeth. Looking at the second and third stages of transmission, although the gear ratio is a fractional number it doesn't follow the criteria of a periodic fractional number. This choice was conscious and it had the aim of simplifying the calculation of the angular position of the intermediate pinion in relation to the respective spline of each intermediate shafts. This matter will be dealt with in the ensuing section 4.1.3.

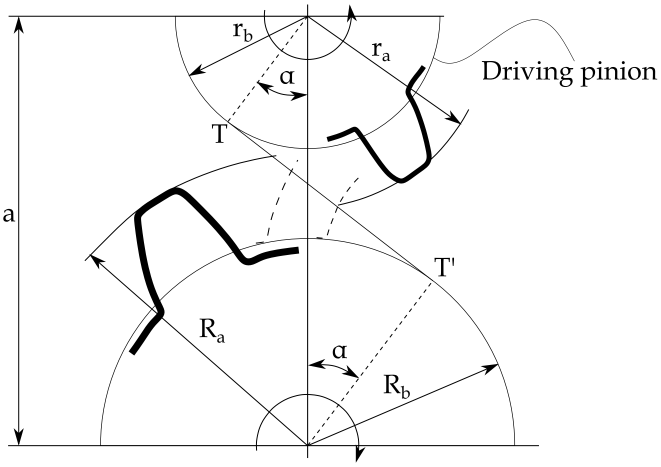


Figure 3.2: Maximum length of the gear meshing line, TT', [31].

Regarding the number of teeth, from figure 3.2 it is verified that the meshing line is limited by the maximum length of TT', and so, consequently, any addendum (tip) circle for which the tangency exceeds this line, is not only useless but also interferes with the root of the tooth with which it will mesh. Given a defined modulus value, equal for each element of the same pair of gears, the addendum circle will be greater when the number of teeth is higher, in the limit tending to infinity, or else considering a gear with a rack. This interference phenomenon continuously occurs during the manufacturing of each gear, being the cutting tool responsible for removing its material from the interference region. This rise to a tooth profile composed of an involute arc and another of trochoid. In this way the non-interference condition is checked; however, it is recommended to ensure a minimum number of teeth so that there is no excessive material removal at the root of the tooth. According to [31], the condition for not occurring this excessive cut is given by,

$$Z_c \geq \frac{2}{\sin^2 \alpha} \tag{3.1}$$

For the most common case, with a pressure angle of 20 °, the minimum number of teeth for no interference is 17 teeth. Important to refer that, for some applications a number of teeth less than 17 is possible, if high profile shifts coefficients is used. Since this gearbox has very high magnitude of forces, such a small number of teeth is not possible to be considered.

Finally, looking at the module gear, it is suggested by [32] that decreasing its value, even though requiring a greater number of teeth, means minimizing the sliding effect in the tooth flanks. Consequently, the meshing efficiency growths. So, this was the premise followed in choosing the module of the different gear stages.

Table 3.1: Number of teeth and module of final gearbox parameters.

Gear stage	**Element**	**No. Teeth**	**Transmission ratio**	**Module** mm
1st	sun	26	5.181	16
	planet	41		
	ring	109		
2nd	gear	120	4.8	8
	pinion	25		
3rd	gear	100	4	5
	pinion	25		

3.1.5 Facewidth

In the design of helical gears, the facewidth is usually based on which it the required load-carrying capacity needed, [26]. In addition, [26] suggest that it is necessary to obtain at least two axial pitches of facewidth, being this value as great as the operating rotational speed.

In fact, the choice of gear facewidth should not be independent of the analysis of its overall size magnitude. This is, an uncontrolled increase in the facewidth of a gear could have heavy costs in its production, since with the increase in facewidth plus the difficulty in ensuring the accuracy of geometry along the entire length of the gear tooth. In order to avoid torsional twist concentrations, the maximum limiting value suggested by [26] for any helical gear pair is equal to its pinion reference diameter. It may cause quite heavy loads at the end of the gear, making impossible to get tooth contact across this much facewidth. For the particular case of a double helical gearing, a face width up to 1.75 times the pinion pitch may be applied. Regarding the bending resistance issues, with both *single* and *double* helical gears, profile teeth modification should be considered in order to minimize its effect. It will be detailed in ensuing section.

Therefore, the facewidth chosen for all toothed elements of this kinematic chain were presented in the Table 3.2. It has to be said that the option of a low *facewidth-axial pitch* ratio for the planetary gearing was intended not only with the above mentioned manufacturing and geometry quality issues, but also with its low operating speed.

Table 3.2: Facewidth, axial pitch and respective ratio of final gearbox parameters.

Gear stage	Element	Face width, b	Nominal module	Axial pitch, p_y	b/m	b/p_y
		mm	mm	mm	-	-
1st	sun	265	16	194.211	16.6	1.4
	planet	256	16.000	1.3		
	ring	265	16.6	1.4		
2nd	gear pinion	188 (40*)	8	59.469	9.25**	2.5***
3rd	gear pinion	127 (25*)	5	37.168	10.2**	2.7***

* intermediate groove of two toothed rim.
** exclusively considered a facewidth corresponding to a single toothed rim.
*** exclusively corresponding to the overall facewidth of the two toothed rim (not considered the intermediate groove).

3.1.6 Profile shift coefficient

Profile shift is commonly used to prevent undercut, balance specific sliding, balance flash temperature, balance bending fatigue life, or even, avoid narrow top lands. Therefore, it is important to define the appropriate driver criteria for this application. According to the fourth part of ISO 61400 standard , for wind turbine gearboxes, which are speed increasers, it is usually best to design the profile shift for balanced specific sliding.

Following [31], it is noted that the wear of the tooth surfaces of the gears increases the greater the specific sliding is and it is more pronounced in the pinion. By following the balanced specific sliding criterium, we are moving toward efficiency by reducing the level of power losses in the several gear meshings.

Profile shift coefficient can be symmetrical, avoiding the variation of the center distance, or, on the contrary, it may cause the axes of their elements to be moved away or approached. Considering a variation of the center distance, the criterion often prevalent was,

$$x_{pinion} + x_{gear} \geq 0. \tag{3.2}$$

However, it may happen that the previous sum takes a negative value suggesting a decrease in the between-axis length, and consequently a considerable decrease in the root tooth thickness of one of the gear pair elements, which decreases it bending strength. After different calculations of planetary gear resistance it was found that the meshing between the sun and the different planet gears was the critical one, in terms of the bending resistance of the root tooth; therefore, with this in mind, results where the sum of profile shift coefficients, between the planet and ring gears, were negative shall be accepted.

Table 3.3: Profile shift coefficient of each gear pair of final gearbox parameters.

Gear stage	**Element**	**Profile shift coefficient, x**	**Sum of prof. shift coef.**
		-	-
1st	sun	0.3934	
	planet	0.3457	0.7391/-0.1786
	ring	-0.5244	
2nd	gear	0.2974	0.3215
	pinion	0.0241	
3rd	gear	0.2964	0.3440
	pinion	0.0476	

3.1.7 Material selection

Given the magnitude of the load to which the transmission group is subjected in wind turbines of this scale, a careful selection of gear materials is required, especially with respect to the ring gear of the planetary stage. The operating pitch diameter of the ring is about 1800 mm with a nominal module of a few tens of millimeters; therefore, there is a restrict list of materials and heat treatments that ensure the necessary strength throughout the overall length and deep of the gear tooth. This allows us to introduce issues such as surface and wear concerns.

Wear is the removal of material due to a mechanical process under conditions of sliding, rolling, or repeated impact between two surfaces into contact. A high increasing of temperature often occurs in the contact zone, allowing the melt of the two surfaces with a consecutive connection material break and material is transferred from one body to the other. This phenomenon is called adhesive wear and some of its causes are: excessive sliding, which is essentially verified in larger parts due to the greater order of magnitude of the tolerances and clearances; high angular acceleration, which is the result of wind gusts; or even, torsional vibrations also characteristic of the operation of these turbines. For all this, the important role of the wear in the design of this gearbox (particularly in larger parts) is evident.

For this, and having in mind that processes of surface heat treatments are used in cases where the concern with the wear resistance prevails over the mechanical resistance, a progressive tooth root flame hardening process was considered for the ring gear. Following the table 3.4, it is the most general shell hardening process applied for gears with this module and diameter.

Table 3.4: Superficial hardening processes according to gear module value, [19].

Process and hardened area	Effect and application	Induction hardening by High frequency	Induction hardening by Med. Frequency	Flame hardening
		Diameter and nominal module (in mm)		Diameter and nominal module (in mm)
Spin hardening exclusive of tooth root	Increase of wear resistance of tooth flanks. Bending fatigue strength not influenced. Permissible bending stress not higher than for normally quenched and tempered parts of the same steel.	Diameter according to generator power; width of gear 10 to100mm; m ≤ 3	Diameter according to generator power; width of gear 35 to 150 mm; in certain cases up to 400 mm; m ≤ 5	Diameter generally up to 450 mm; with special machines larger; m ≤ 6 in special cases m ≤ 12
Spin hardening including tooth root	Increase of wear resistance of tooth flanks and bending fatigue strength in the tooth root. Permissible bending stress 30 to 50% higher than for normally quenched and tempered. Partly replaces case-hardened gears.	Diameter according to generator power; width of gear 10 to100mm; m ≤ 5	Diameter according to generator power; width of gear 35 to 150 mm; in certain cases up to 400 mm; m ≤ 5	Diameter up to 450 mm; m ≤ 6 in special cases m ≤ 10
Progressive hardening of both flanks	Increase of wear resistance of tooth flanks. Bending fatigue strength not influenced. Permissible bending stress not higher than for normally quenched and tempered parts of the same steel.	Diameter unlimited m ≥ 2	Diameter unlimited m ≥ 5	Diameter unlimited m ≥ 6
Progressive tooth root hardening	Increase of wear resistance of tooth flanks and bending fatigue strength in the tooth root. Permissible bending stress 30 to 50% higher than for normally quenched and tempered. Partly replaces case-hardened gears.	Diameter unlimited m ≥ 2	Diameter unlimited m = 5 to 30	Diameter unlimited m ≥ 10

Regarding the material strength and following the first part of IEC 61400 standard, a MQ material quality shall be verified. If we cross this information with the gear material data base of KISSsoft® software, the following list of possible materials (according to the DIN designation) shows up,

- 34 CrNiMo 6;
- 36 CrNiMo 4;
- 42 CrMo 4.

Knowing that Nickel is a noble material with a high cost associated, 34 CrNiMo 6 and 36 CrNiMo 4 were rejected. So, **42 CrMo 4** was the selected one for the ring gear material. Its yield point shown in the table 3.5 also respect the typical range values of bending stresses —[345; 483]— on wind turbine gearboxes, referred by [15]. In that way we ensure that the ring gear works in the plastic domain.

Table 3.5: 42 CrMo 4 (DIN) mechanical properties, [33].

Yield strength MPa	**Tensile strength** Mpa
≥ 500	≥ 755

Contrary to what happens with the ring, it is important that the core of the sun gear, which will be manufactured directly on the shaft and hence have to allow low deflection, to provide high yield strength. Regarding the planet gears, which shall house its tapered roller bearing, a bore surface hardness at least of 55 HRC is suggested by the WT standard IEC 61400. Bearing this in mind and considering the high strength and hardenability that were required, the material chosen was the **18 CrNiMo 7-6** case-carburized and -hardened (tab. 3.6). The same for the all others gears.

Table 3.6: 18 CrNiMo 7-6 (DIN) mechanical properties, [33].

Yield strength MPa	**Tensile strength** Mpa
≥ 735	≥ 885

It is important to state that following the material selection drivers of ISO 61400-1, the material strength values shall be at least correspondent to the MQ quality. Wind turbine gears require smooth tooth surfaces to ensure adequate load capacity. Smooth surfaces are especially important with regard to micropitting resistance. Roughly, for external gears maximum flank surface roughness shall be Ra = 0,8 µm. Maximum flank surface roughness for internal gears shall be Ra = 1,6 µm. IEC 61400-4 standard suggest a specific surface roughness level for each gear stage. It is shown in the Table 3.7 and it was considered in the KISSsoft® calculation.

Table 3.7: Recommended gear tooth surface roughness, [34].

Gear	$\mathbf{R_a}$ (μm)
high speed pinion and gear	≤ 0.7
intermediate pinion and gear	≤ 0.7
low speed pinion and gear	≤ 0.6
low speed sun and planet	≤ 0.5

3.1.8 Input and Output conditions

3.1.8.1 General conditions

With the information so far, it is possible to set the wind turbine gearbox input and operation conditions.

Since the gearbox have a input power of **2.0 MW**, having in mind what was said in Section 2.4.1, an input speed of **15 rpm** is chosen as the nominal input speed of the gearbox. Notice that, according to the Table 2.5, this is near to the market rates of the wind turbine gearboxes industry.

Typical generators for *gearbox-generator* wind turbine have 4 poles. Bearing this in mind, according to the Equation (2.1) for a **4 pole**, **50 Hz** generator, a output speed of **1500 rpm** is needed, which is provided by a overall transmission ratio[2] of about **1:100**.

As stated in Section 2.4.1, a wind turbine gearbox normally have multiple stages to achieve the high conversion ratio. So, supported by the research presented in Section 2.4.3, a ***one planetary/two parallel*** stages has been chosen. It is widely used for a rated power wind turbine about 2.0 MW. For the transmission ratio of each stage, a 7 x 4 x 3.6 [3] group of transmission ratio was proposed for this design.

In fact, a decreasing of each stage transmission ratio is typically verified along the transmission chain. However, high transmission ratio imply high dimensional pinion variation in relation to the gear size, compromising the objective of a compact design as possible. Given the first planetary stage, the magnitude of size will be essentially defined by the external diameter of the ring, so a more uniform distribution of the single transmission ratios should be studied. The selected one was the **5.2 x 4.8 x 4** group of transmission ratios. This change can determine a significant reduction of the outer gearbox diameter. So, a detailed comparison between the first purposed group of transmission ratio and the last was made.

3.1.8.2 Operating temperature

Considering the operating conditions presented in Section 2.4.1, a **40 ˚C** ambient temperature is considered. This decision was to avoid considering nominal operating conditions close to the limit conditions, but without making a too strict and inflexible choice bringing excessive oversizing of the WT gearbox.

3.1.8.3 Required life time

The design life time of a wind turbine is defined, in the first part of IEC 61400, as at least 20 years, so it is expected to ensure that gearbox have at least the same durability. Considering that, 175 200 hours of service was regarded as its required life time.

As it will be shown in following chapters, it was considered for gearbox optimization a design load cases that, contrarily what happens in above sentence, took into account

[2] also known as *global transmission ratio*.

[3] each of the values represent the multiplication factor of the speed, per stage.

particular load cases like parked time. It embody the requirements for loads resulting from atmospheric turbulence that occurs during normal operation of a wind turbine throughout its lifetime. DLC also embodies the requirements for ultimate loading resulting from extreme turbulence conditions, specific transient cases that have been selected as potentially critical events in the life of a wind turbine.

Regarding the rolling bearings, its design shall consider the expected amount of rotation during gearbox lifetime. Modified reference rating life L_{nmrh} shall be calculated in accordance with ISO/TS 16281 and it shall meet or exceed the specified design lifetime for the gearbox (175 200 hours). Notice that IEC 61400 recommends to considered a permissible failure probability equal or less than 10 percent. So, when setting the rolling bearing life time it should not be forgotten that a 10 percent of failure probability was considered.

3.1.9 Load factors

3.1.9.1 Application factor, K_A

One of the input parameters of KISSsoft® calculation tool is the application factor K_A. It classifies the operational behavior of driver and driven elements of the system according to the following table,

Table 3.8: Assignment of operational behavior to application factor. Adapted from the KISSsof® software instructions.

Operating behavior of the driving machine	**Operational behavior of the driven machine**			
	uniform	**moderate shocks**	**medium**	**heavy shocks**
uniform	1.00	1.25	1.50	1.75
ligth shocks	1.10	1.35	1.60	1.85
moderate shocks	1.25	1.50	1.75	2.00
heavy shocks	1.50	1.75	2.00	2.25

Considering that the rotor turbine occasionally suffers peaks of loads resulting, for example, from wind bursts and bearing in mind a continuous operation of the gearbox the chosen application factor was,

$$K_A = 1.25. \tag{3.3}$$

3.1.9.2 Dynamic factor, K_V

The dynamic facto K_V considers the load increase on the flank and on the root because of toothing rigidity variations during meshing. Should the frequency of the rigidity variations/number of revolutions be directly comparable to the meshing inherent frequency, so will appear dynamic additional stresses, [18]. Wind turbine standard of IEC mentions that if the calculation indicates a dynamic factor less than 1.05, a K_V of 1.05 shall be considered as a minimum value.

It is known that the dynamic factor has a significantly effect about gear rating, and after a few gearing calculation it was detected that without make any value restrictions, a K_V value less than the recommended one always occur. So, having verified that, the following dynamic factor was set,

$$K_V = 1.05. \tag{3.4}$$

3.1.9.3 Mesh load factor, K_γ

Due to manufacturing tolerances and deformations, the load distribution between the load paths (Planet gears) is not equally balanced; therefore, for the calculation of planetary sets, the load has to be multiplied by a K_γ factor, [18]. The data values for WT strongly differ according to each source, see Figure 3.3.

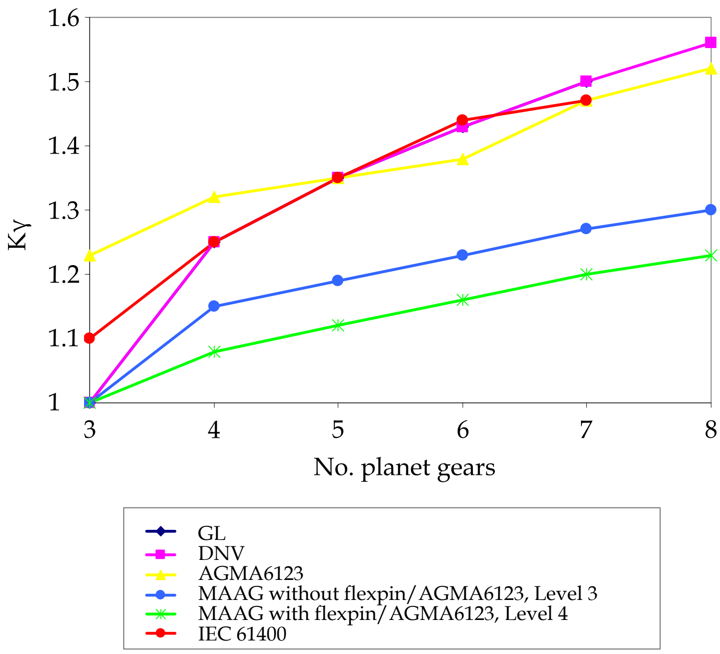


Figure 3.3: Load distribution factor as a function of the number of planet gears, [18].

Like for all the others safety coefficients decision made, for the mesh load factor the IEC standard was followed. So regarding the planetary stage, the guidance values of the mesh factor suggested for them is shown in the Table 3.9. These values are depending on the number of planets.

Table 3.9: Mesh Load Factor K_γ for Planetary Stages, [34].

Number of planets	3	4	5	6	7
Mesh load factor, K_γ	1.10	1.25	1.35	1.44	1.47

Since a three planet gear configuration was chosen in previous sections, the following mesh load factor was set,

$$K_\gamma = 1.10. \tag{3.5}$$

Although the second and third stages of speed increase are not planetary gearing, power slip also occurs in these transmission stages, having multiple paths for forces distribution;

so, the factor K_γ chosen for the first stage was also applied in the last stages of transmission.

3.1.9.4 Load Distribution Factors, $K_{H\alpha}$ and $K_{H\beta}$

The transverse factor $K_{H\alpha}$ considers the effects of an uneven load increase on the flank due to pitch errors and irregular load distribution over several teeth in meshing. According to [34], because of the toothing qualities required for wind turbine gearboxes, a unitary load distribution factor $K_{H\alpha}$ may be considered.

The effect of meshing force distribution over the toothing width upon the flank loads are described by the face load factor $K_{H\beta}$. In addition to this extreme loads, the influence of production variation on shaft parallelism and tooth alignment of pinion and gear should be included in the value of mesh misalignment, and included in the $K_{H\beta}$ definition.

Load distribution factors recommended are presented in the following table.

Table 3.10: Pitting resistance, bending strength and scuffing resistance safety factors for the final gearbox solution, [34].

Transverse load factor $K_{H\alpha}$	**Face load factor $K_{H\beta}$**
1.00	1.15

3.1.10 Safety factors

Root safety, flank safety and safety against scuffing at flash temperature were the criteria took into account for gear sizing, service life time, and later, for load spectrum calculations.

In earlier times, the mechanical designer tried to calculate a “safe” stress. It was thought that all parts with less than the safe stress limit would perform without failure, [26]. In fact, mechanical components, such as gears, have a probability of failure. In order to minimize it, several safety coefficients should be considered, defining a safe stress range in which the gear can operate with a low probability of failure.

The failure modes of gears include plastic fracture and/or deformation of the teeth and/or gear body, damage to the surface of the teeth, and corrosion, [35]. Typically, fracture occurs at the root of the tooth, in fatigue processes caused by many load cycles, where it may be possible to identify fatigue cracks at the fracture surface. There may also be fractures originating from the body of the gear (eg, from the spline feature). Flank defects include pitting and scuffing due to localized micro-welding effects.

In order to minimize the probability of the aforementioned rupture, the following safety factors was considered as the minimum recommended (tab. 3.11).

3.1.11 Lubricant and lubricating method

3.1.11.1 Lubricating oil selection

Lubrication is one of the most delicate issues in the operation of a gear. It is known that in the gear meshing process, great thermal and pressure loads can occur on the teeth in

Table 3.11: Pitting resistance, bending strength and scuffing resistance safety factors for the final gearbox solution, [34].

Safety factor for for pitting resistance $\mathbf{S_H}$	**Safety factor for bending strength $\mathbf{S_F}$**	**Safety factor for for scuffing resistance $\mathbf{S_B}$**
1.25	1.56	1.3

contact, may even cause them to melt and, consequently, the breakage of the mechanism. Thus, through several lubrication methods it is possible to form a lubricant film which separates the surfaces by avoiding metal-to-metal joint. This will obviously contribute to a decrease in the friction coefficient between the surfaces in contact.As mentioned above, there is a thermal effect associated with the contact of the teeth during the engagement. Also here the lubricating film seems indispensable in the way that it provide the evacuation of the heat generated. Like that, the materials ruin of the operating gear pair, at high temperatures, is prevented.

It is extremely important to clarify that the three main conditions for lubrication are the following:

- geometry (convergent shape);
- speed;
- viscosity (body).

In other words, and very briefly, when velocity-excited, the lubricant is guided to a zone of convergence (the beginning zone of the meshing), suffering a smashing effect and, therefore, the formation of a lubricating film which separate both teeth flanks in contact.

The choice of the type of lubricant takes into account specifications such as:

- transmitted power;
- operating and start-up temperatures;
- gear type;
- gear speed and transmission ratio;
- load characteristics;
- expected life-time;
- application method.

These factors lead to the determination of the physical, chemical and operational properties of the oil to be used, whose properties are closely related to multiple constrains like kinematic and dynamic viscosity, viscosity index, pour point, additives and costs.

Given the range of low speed and high torque of the wind turbine gearbox, the IEC 61400-4 standard recommends the use of a mineral or synthetic oil base. Although mineral oils are less expensive, synthetic oils have lower pour point which means lower feed temperature and thus lower energy costs associated with cold start preheating mechanisms. In addition, synthetic lubricants have a high viscosity index; so, they table a higher thermal inertia than mineral oils, providing a less significant temperature-properties (e.g. viscosity) variation or, in other words, a more stability of the lubricant behavior.

In a gearbox it is advantageous to operate at temperatures as low as possible so that the integrity of the components is not compromised; thus, once again, the synthetic oils show to be the best option since they allow the operation of the gears at lower temperatures than those obtained with oils of mineral base. Low operating temperatures ensure longer gear and rolling bearing life due to the increased film thickness. It also provide a higher life estimate for the lubricant by decreasing the oxidation effect, which is extremely important due to the long life expected for the wind turbine conversion system.

Table 3.12: Characteristics of the *Optigear^{TM} Synthetic 1710*, *Optigear^{TM} Synthetic X* and *Optigear^{TM} Synthetic A* oils.

Parameter	**Unit**	**Value**		
Oil family		**Synthetic 1710**	**Synthetic X**	**Synthetic A**
Density @ 15 ˚C	**kg/m^3**	875	**822**	870
Kin. viscosity @ 40 ˚C	**mm^2/s**	320	**325**	330
Kin. viscosity @ 100 ˚C	**mm^2/s**	31.2	**34.9**	33
Viscosity index		138	**152**	140
Flash point	**˚C**	240	**250**	220
Pour point	**˚C**	-30	**-33**	-36

In order to fulfil the needs of planetary gears stages, it is general practice to use **ISO VG 320** grade lubricants for wind turbine multiplication gearbox, [36]. So, after a search in the manufacturers' catalogs, the three synthetic oil families was chosen to compare: *Optigear^{TM} Synthetic 1710*, *Optigear^{TM} Synthetic X* and *Optigear^{TM} Synthetic A*, table 3.12.

Regarding the gear type, helical gear and double helical gear have a sliding component along the line of contact, which is result of the helix angle. This has little or no effect on the film in the contact area; however,in the convergent zone the sliding component tends to wipe the lubricant sideways. For that, not as much lubricant is available to be drawn into the contact area, being the resultant pressure increase in the convergent zone rather lower. These effects may contribute to a need for slightly higher viscosity lubricants. With the *Synthetic X* and *Synthetic A* having higher and similar viscosity levels, both shows as candidate oils for this gearbox.

Transmission ratio influences the selection of lubricating oil as long as the higher its value the more gears stages must exist. The low speed gear in the gear set is usually the most critical in the formation of the EHD film, [37]. At low gear speeds more time is available for oil to be squeezed from the contact area and less oil is drawn into the convergent zone. For that, higher viscosity oil are required. Here, *Synthetic X* oil is preferred since, at higher temperatures, it has higher viscosity values than other ones do. The higher speed of the last gear stage provide higher sliding and rolling speed of individual teeth. To help reduce the fiction effect resultant of that, a EP additive are recommended.

Gears design for higher power ratings will have wider teeth, teeth of larger cross section (or both), which shall cause mechanical and fluid friction to be greater. As a result, larger gear sets, transmitting more power, tend to run hotter. Considering this severe operating conditions, the use of additives is a necessary practice. Ideally it would be beneficial to use lubricant compositions adapted to the needs of each stage — extreme pressure additives (EP) for high-speed stages and anti wear additives (AW) for high torque stages — but this is not an efficient practice in way it would imply a sealed sectioning of the different stages in order to avoid contamination. Therefore, a compromise between the two types of additives is achieved with the use of this Castrol Optigear oil.

It should be noted that, the nature of the load on any gear set also have an importance in oil lubricating selection. In some operations, the conditions may be more severe owing to overloads or to a combination of heavy loads and extreme shock loads. Despite of being considered during the calculation process (like presented in ensuing section 4.1.1), this condition generally requires the use of extreme pressure (EP) oils.

In summary, starting with the oil base and bearing in mind the advantages of using fully synthetic base oils, *synthetic 1710* is the first that was excluded due to its mineral/PAO oil base. It is also possible to choose between *Synthetic X* and *Synthetic A*; however, based on all the conditions aforementioned, the lubricant chosen was the **Castrol Optigear Synthetic X 320**. This, as stated by the manufacturer, is ideal for gearboxes with planetary gear temperatures between -35 °C and + 90 °C[4], [38]. The fully synthetic polyalphaolefine base oils have a significant role in this type of application due to the excellent viscosity-temperature behavior provided by the high viscosity index, low coefficient of friction and good miscibility with additives, [36].

3.1.11.2 Pressure viscosity parameters

The factors k and s are used to determine the pressure viscosity coefficient, α. The k value is a linear multiple, while the s value is an exponential power for the dynamic viscosity, μ. These coefficients were checked and applied in accordance with the AGMA 925-A03 [39] standard, and they are presented in the following table.

Table 3.13: Data for determining pressure–viscosity coefficient (acc. to AGMA 925-A03 [39]): k and s factors.

Lubricant	**ISO VG grade**	k	s
PAO	320	0.010326	0.0507

3.1.11.3 Scuffing and micropitting tests parameters

Scuffing occurs locally where the gears are in mesh, i.e. where the roughness peaks in contact temperatures rise sharply (*flash temperatures*), depending on the load, peripheral

[4]According to standard [34], selection of the correct viscosity grade for the gearbox must be based on the operating conditions and not on the start-up ones; and, therefore, operating temperature range — from -29 °C to +50 °C — proposed by [15] must not be forgotten.

speed and oil sump temperature. At these contact points, the surfaces weld together briefly and are torn apart again as the gears revolve, which leads to partial destruction of the surfaces, [40]. FZG test rig is a machinery system that allows to determinate the extent to which gear lubricants help to prevent scuffing on the tooth faces. Bearing in mind what is stated in [34], the scuffing capacity of the oil shall be determined by the scuffing test FZG A/8.3/90 according to ISO 14635-1. Following this test method, one stage lower than the fail load stage shall be used for the scuffing analysis. Having that in mind and considering that the lowest class of scuffing failure stated for [34] is the twelfth one, class **11** can be used in scuffing analysis and it was the one considered in KISSsoft® calculation.

Regarding micropitting phenomenon, this failure consists of very small cracks and pores on the surface of tooth flanks. Micropitting looks greyish and causes material loss and a change in the profile form of the tooth flanks, which can lead to pitting and breakdown of the gears, [41].

Again following [34], one of the test which may perform for to analysis and set the micropitting resistance class is the test gears type C-GF run at a circumferential speed of 8.3 m/s and a lubricant temperature of 60 °C or 90 °C. In order to ensure a few difference between the temperature of the lubricant (65 °C as stated in ensuing section) and the temperature used when the micropitting test process performs, it was selected a temperature FZG test of **60 °C**. The load stage applied was the **10th** ones by suggestion of IEC wind turbine standard.

Table 3.14: Test methods for lubricant performance.

Property	Test method	Test conditions	Requirement
Scuffing	ISO 14635-1	A/8.3/90	11
Micropitting	FVA 54/I-IV	CGF/8.3/60	10

3.1.11.4 Lubricating method selection

Oil bath lubrication and injected lubrication are the most common lubrication methods for gearboxes. By making a direct comparison of both, in terms of speed and size as suggested by the Figure 3.4, we find that the points of influence are a few dispersed. Here, the configuration of the different transmission stages and the dissipation of the heat generated are determinative factors in the final choice. The choice must be closely connected to gearbox application. In case of *low speed-high torque* gearboxes such as wind turbines ones, where the thermal load associated with high stresses on the teeth flanks is high, oil-bath lubrication may be insufficient, being rejected to the injection method. So, a **65 °C oil injection** as the one selected for this work. In this oil lubrication method, the oil is injected with pressure to the surroundings of the beginning of the gear meshing zone; then, it is “pushed” into the contact zone by the action of the gear motion, [42]. The direction of the oil jet shall be such that it allows both flanks impregnation, ensuring oil abundance in the ensuing contact zone and, in this way, to avoid a lack of access to the oil in the formation of the lubricating film.

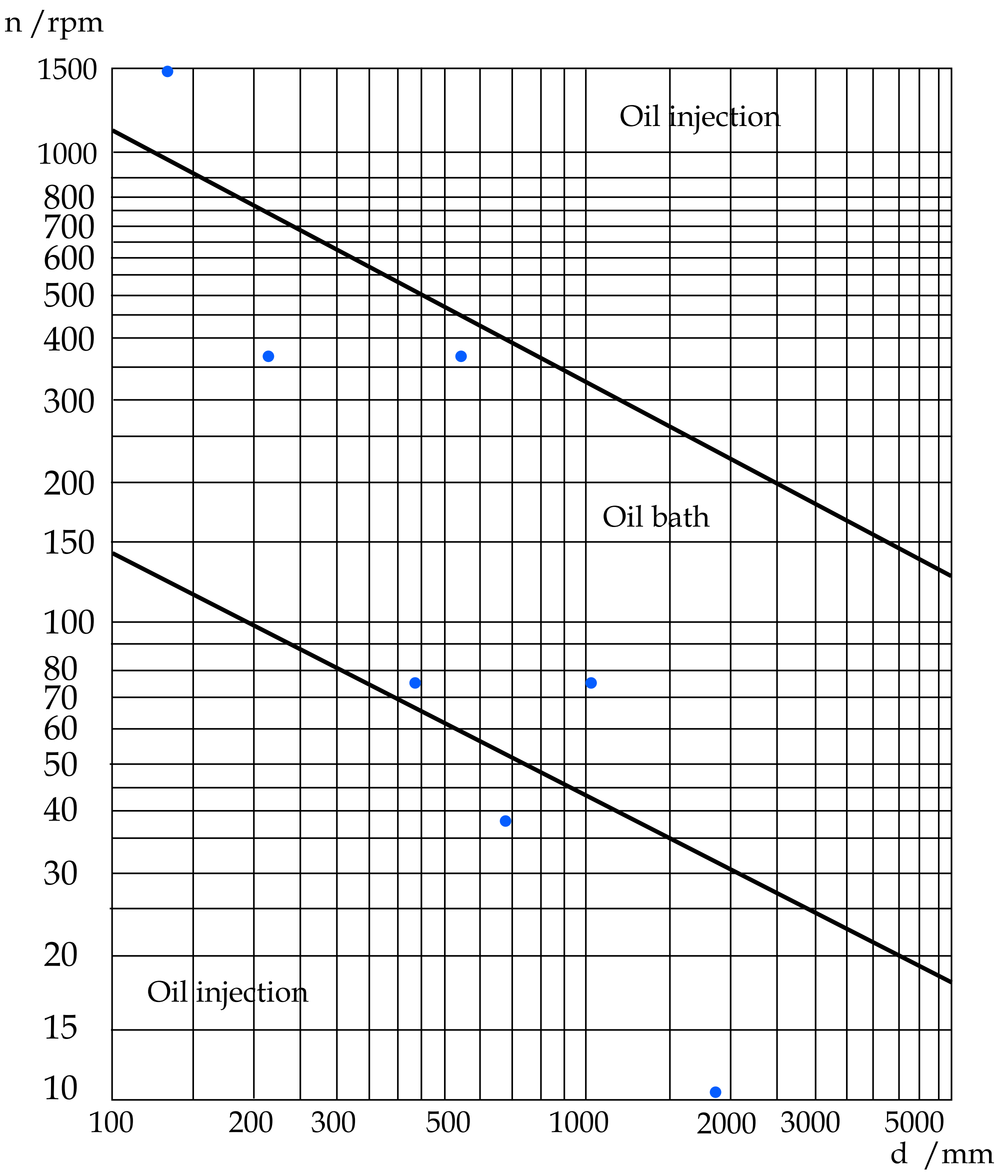


Figure 3.4: Lubrication methods by Henriot, [42].

3.1.12 Results

Table 3.15: First gear stage dimensioning results from KISSsoft® regarding the nominal rated power.

		Sun	Planets	Ring
Power	**kW**		2000	
Speed	**rpm**	77.9	39.9	0
Speed planet carrier	**rpm**		15	
Number of teeth	**-**	26	41	109
Normal module	**mm**		16	
Overall transmission ratio	**-**		5.18:1	
Center distance	**mm**		566	
Facewidth	**mm**	265	256	265
Pressure angle at normal section	^{o}		20	
Helix angle at ref. circle	^{o}		15	
Accuracy grade	**ISO1328:1995**	6	6	7
Reference diameter	**mm**	430.7	679.1	1805.5
Tip diameter	**mm**	473.8	720.7	1790.3
Material	**DIN**	18 CrNiMo 7-6		42 CrMo 4
Heat treatment		Case-carburized		Flame hard.
Surface hardness	**HRC**	61		56
Profile shift coefficient	**-**	+0.3934	+0.3457	-0.5244
Sum of profile shift coef.	**-**	+0.7391		-0.1786
Specific sliding at the tip	**-**	0.488	0.488/0.167	0.195
Specific sliding at the root	**-**	-0.952	-0.952/-0.242	-0.201
Circumf. speed at tip circle	**m/s**	1.506	1.933	0
Transverse contact ratio, ϵ_α	**-**	1.424		1.650
Overlap ratio, ϵ_β	**-**	1.318		1.318
Total contact ratio	**-**	2.742		2.968
Safety f/ tooth root stress*, $\mathbf{S_F}$	**-**	2.12	1.44/ 1.72	2.06
Safety f/ pitting resistance* , $\mathbf{S_H}$	**-**	1.26	1.32/ 2.79	2.26
Safety f/ scuffing resistance* , $\mathbf{S_B}$	**-**	3.746		28.909
Mass	**kg**	313.397	739.722	924.625
Total mass	**kg**		3457.187	
Mean friction coef. by Niemann	**-**	0.041		0.029
Wear sliding coef. by Niemann	**-**	0.694		0.302
Gear power loss	**kW**	2.680		0.573
Total power loss	**kW**		9.760	
Total efficiency	**%**		99.5	

* Safety coefficient according to the nominal load, without considering any load spectrum. The final solution can be consulted in Section 4.1.1 of the optimization chapter.

Table 3.16: Second gear stage dimensioning results from KISSsoft® regarding the nominal rated power.

		Gear	Pinion
Power*	**kW**	1000	
Speed	**rpm**	77.9	373.8
Number of teeth	**-**	120	25
Normal module	**mm**	8	
Overall transmission ratio	**-**	4.8:1	
Center distance	**mm**	642.500	
Total facewidth of gear	**mm**	188	
Width of intermediate groove	**mm**	40	
Facewidth for calculation	**mm**	74	
Pressure angle at normal section	^{o}	20	
Helix angle at ref. circle	^{o}	25	
Accuracy grade	**ISO1328:1995**	6	
Reference diameter	**mm**	1059.2	220.7
Tip diameter	**mm**	1075.6	241.4
Material	**DIN**	18 CrNiMo 7-6	
Heat treatment		Case-carburized	
Surface hardness	**HRC**	61	
Profile shift coefficient	**-**	+0.0241	+0.2974
Sum of profile shift coef.	**-**	+0.3215	
Specific sliding at the tip	**-**	0.409	0.409
Specific sliding at the root	**-**	-0.691	-0.691
Circumf. speed at tip circle	**m/s**	4.387	4.724
Transverse contact ratio, ϵ_α	**-**	1.437	
Overlap ratio, ϵ_β	**-**	1.244	
Total contact ratio	**-**	2.682	
Safety f/ tooth root stress, $\mathbf{S_F}$**	**-**	1.51	1.64
Safety f/ pitting resistance , $\mathbf{S_H}$**	**-**	1.27	1.27
Safety f/ scuffing resistance , $\mathbf{S_B}$**	**-**	2.980	
Mass	**kg**	1293.154	57.701
Total mass***	**kg**	1408.56	
Mean friction coef. by Niemann	**-**	0.044	
Wear sliding coef. by Niemann	**-**	0.587	
Gear power loss	**kW**	4.537	
Total power loss	**kW**	9.074	
Total efficiency	**%**	99.546	

* Rated power for each single power path of the mentioned double branch stage.
** Safety coefficient according to the nominal load, without considering any load spectrum. The final solution can be consulted in Section 4.1.1 of the optimization chapter.
*** Total mass including the gear and both pinion.

Table 3.17: Third gear stage dimensioning results from KISSsoft® regarding the nominal rated power.

		Gear	Pinion
Power*	**kW**	1000	
Speed	**rpm**	373.8	1495.4
Number of teeth	**-**	100	25
Normal module	**mm**	5	
Overall transmission ratio	**-**	4:1	
Center distance	**mm**	346.500	
Total facewidth of gear	**mm**	127	
Width of intermediate groove	**mm**	25	
Facewidth for calculation	**mm**	51	
Pressure angle at normal section	^{o}	20	
Helix angle at ref. circle	^{o}	25	
Accuracy grade	**ISO1328:1995**	6	
Reference diameter	**mm**	551.7	137.9
Tip diameter	**mm**	562.1	150.8
Material	**DIN**	18 CrNiMo 7-6	
Heat treatment		Case-carburized	
Surface hardness	**HRC**	61	
Profile shift coefficient	**-**	+0.0476	+0.2964
Sum of profile shift coef.	**-**	+0.3440	
Specific sliding at the tip	**-**	0.417	0.417
Specific sliding at the root	**-**	-0.715	-0.715
Circumf. speed at tip circle	**m/s**	11.002	11.810
Transverse contact ratio, ϵ_α	**-**	1.430	
Overlap ratio, ϵ_β	**-**	1.372	
Total contact ratio	**-**	2.802	
Safety f/ tooth root stress, S_F**	**-**	1.67	1.80
Safety f/ pitting resistance , S_H**	**-**	1.69	
Safety f/ scuffing resistance , S_B**	**-**	3.089	
Mass	**kg**	237.020	15.223
Total mass***	**kg**	267.466	
Mean friction coef. by Niemann	**-**	0.041	
Wear sliding coef. by Niemann	**-**	0.596	
Gear power loss	**kW**	4.300	
Total power loss	**kW**	8.600	
Total efficiency	**%**	99.570	

* Rated power for each single power path of the mentioned double branch stage.
** Safety coefficient according to the nominal load, without considering any load spectrum. The final solution can be consulted in Section 4.1.1 of the optimization chapter.
*** Total mass including the gear and both pinion.

Chapter 4

Gearbox Optimization

In this chapter, the final solution of the wind turbine gearbox is described. Here are important and essential details that define the improved performance version of the machine that was presented in the prior chapter. The changes are basically related with the increase of the root teeth bending resistance, the development of the gearbox assembly, the refinement of the material choices, and the presentation of the lubricant selection as well as the gearbox sealing mechanisms.

4.1 Gears

As suggested by the name, the most important machine elements of the gearbox are **the gears**; and, therefore a in-depth study for improvement should be done. In other words, using the KISSsoft® calculation tool, several combination of meshing gear parameters was tried until the best and most compact one stands out. After defining the ultimate gearing solution, there are other small changes that must be considered for their (the gears) optimization:

- selection of an adequate gear material;
- add on of a tooth profile modification;
- selection of an adequate oil lubricant (suited for the application);
- reduction of the gear mass;
- selection of an appropriate gear attachment configuration.

Each of these modifications will be discussed and applied as much as possible. It should be noted that other assembly issues, in particular about the double branch stages, will also be shown.

4.1.1 Load spectrum application

In a macro sense, gradients of temperature, which are the result of the solar heating, create movements of air masses that originate the wind. In fact, the atmosphere layers closest to the earth's surface are more irradiated than the next ones. So, its warmer and lighter air rises to the outer layers of the atmosphere and is replaced by a return cooler flow of air coming from the next atmosphere layers. This air circulation is also affected by the Coriolis forces associated with the rotation of the Earth. The result from an exact balance between this Coriolis force and the pressure gradient force is named as the geostrophic wind. It is directed parallel to isobars (lines of constant pressure at a given height).

In lower layers of the atmosphere, winds are delayed by frictional forces and obstacles altering not only their speed but also their direction, originating turbulent flows. The presence of seas causes extra air masses circulation. The combination of this both effects provide wind speed variations over a wide range of amplitudes. All these air movements are called local winds.

From the above and knowing that the wind result from the combination of the geostrophic and local winds, we can state that it depends on multiple factors like site location, ambient conditions, working height above the ground level, roughness of the terrain and possible surrounding obstacles. It is also important to note that disturbances in any of these conditions will cause disturbances in the turbine load conditions. Thus, in order to achieve a closer approximation of the real wind turbine operation, a load spectrum was considered. Load distributions may be expressed in time-at-level, being each bin the time duration that the gearbox are under this load. In addition to the operating conditions of the power production , the load spectrum should contain extreme and/or particular conditions like start up, braking procedures, idling and stopped time. The load spectrum durations may not add up to the design life even when idling is included. For this work, it was considered a load spectrum suggested by the IEC 61400 standard, represented in Figure 4.1.

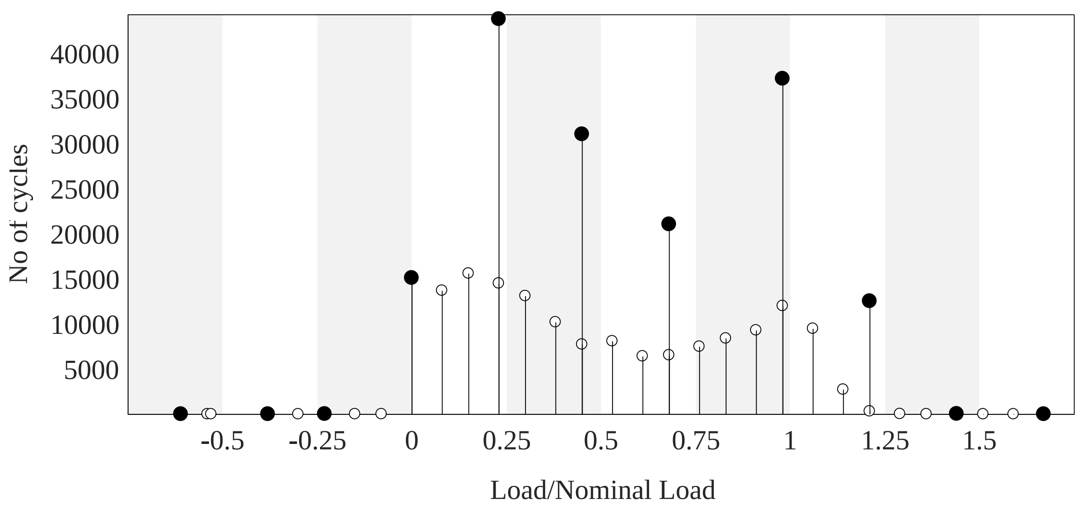


Figure 4.1: Revolutions distribution. Adapted from a DLC suggested by IEC 61400-4, [34].

The load spectrum presented in IEC 61400 standard is detailed in 16 different bins. The high number of short time intervals that IEC 61400 proposes has a high computational cost, in which the higher the number of bins the more time consuming is the calculation with the KISSsoft tool. Thus, a spectrum simplification strategy was applied. It is based on the distribution of the various single bins by different groups of load levels. From one group to the next, there is an increase in the load factor of +0.25, and the overall load range considered is between -0.75T_{nom} and 1.75T_{nom}. For a better understanding, all single bins with a load factor between -0.75 and -0.5 incorporate the first load group, then, the load factor levels between -0.5 and -0.25 belong to the second load group and so on. Each one of the groups has one main bin that represents it. Each one of the main bin is associated with a load factor equal to the highest load factor between the single bins contained therein, and it has a duration that is equal to the sum of all its single bins duration. We are saying that the wind turbine will perform all the cycles of each group at its highest load level, and consequently, dimensioning the gearbox for the most critical situations.

Table 4.1: Safety comparison between nominal power dimensioned gearing and a load spectrum idem.

		Nominal load			**Load spectrum**	
Stage	$\mathbf{S_F}$	$\mathbf{S_H}$	$\mathbf{S_B}$	$\mathbf{S_F}$	$\mathbf{S_H}$	$\mathbf{S_B}$
1st	1.440	1.260	3.756	1.566	1.380	2.845
2nd	1.510	1.270	3.980	1.560	1.286	1.918
3rd	1.560	1.300	3.089	1.566	1.321	1.995

4.1.2 Profile teeth modifications

Due to errors of manufacture, deflections of mountings under load, and deflections of the teeth under load, it is often found that the gear teeth do not perform as they are assumed according to theory. As a result, premature contact at the tips of the teeth occurs, giving rise to noise and/or gear failures, Figure 4.2. To reduce these causes of excessive tooth loads, profile modification, is a usual practice.

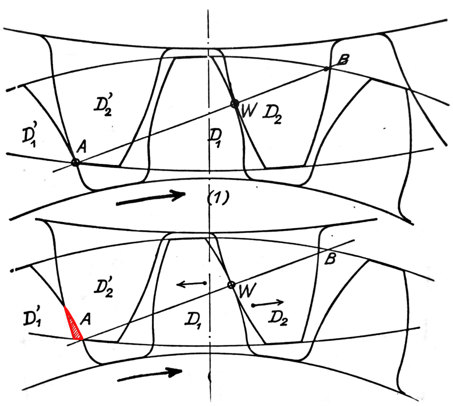


Figure 4.2: Deformation and meshing interference in loaded, pure involute gear, [43].

For better understanding, the Figure 4.2 shows a gear meshing in which two teeth, D_1 and D_2, are in contact at the point W. D_1' and D_2' represent the following pair of teeth to be meshed and so on. Disregarding manufacturing errors, when the teeth are not loaded, a normal (general) contact of the teeth in A would occur. On the contrary, when the teeth are loaded, a bending effect occurs causing interference between the tip of D'_2 and the root of D'_1. This phenomenon is obviously a source of noise but not only. Due to the abrupt intervention of D'_2 on D'_1, the lubricant film fail, questioning the resistance of the surfaces which may cause propagation of wear problems on the flanks of the teeth. Specially in cases of heavily loaded gears, such as those, it is of great interest to smooth the contact between pinions and gears. Tip relief is a way of that. KISSsoft® allows various profile modifications; however, in this work, only tip relief will be analysed.

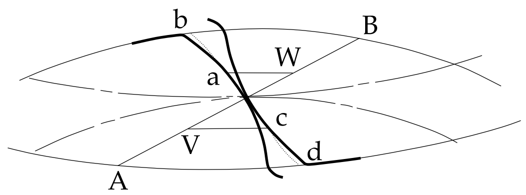


Figure 4.3: Material removal region in a tip relief modification, [43].

Briefly explaining, the tip relief consists of removing a very small quantity of material from the teeth's tip. Following figure 4.3, the removal of material must begin at a for the tooth D_1 and at c for the tooth D_2. In the distance comprised between these markers only one pair of teeth is in contact and at a single point. Thus, the input quantification parameters of the tip relief in the KISSsoft® calculation tool are represented in figure 4.4, where C_a is the tip relief coefficient (or by default just *tip relief*) and X_{C_a} is the tip relief factor 1 (or by default just *tip relief factor*). Tip relief factor defines the quotient from the calculated tip relief length L_{C_a} and normal module m_n.

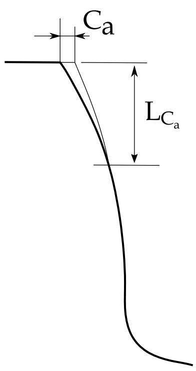


Figure 4.4: Tip relief on a gear tooth, [43].

In figure 4.5 can be observed the effect that tip relief has on the load distribution along of the meshing line. Namely, with this tooth profile modification a more uniform and continuous load is achieved, along the meshing line, resulting into a more adequate hertz pressure (σ_H) distribution and in a lower maximum value of it.

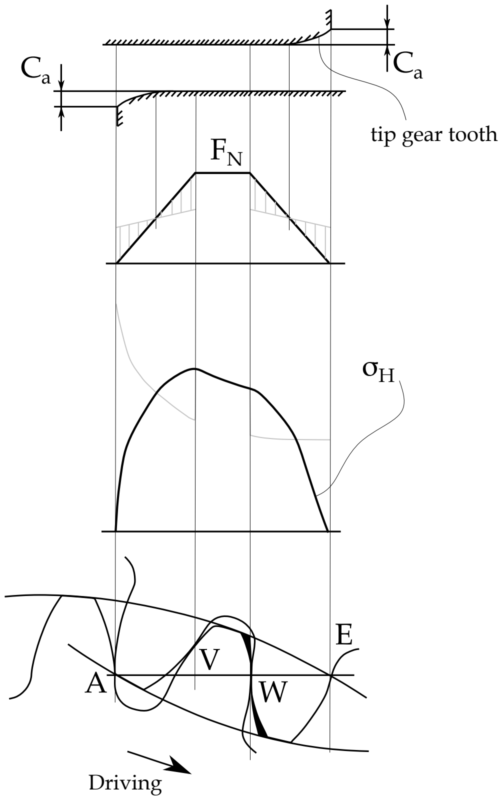


Figure 4.5: Profile diagram of the driving flank. Profile diagram of the driven flank. Load variation. Hertz pressure profile variation. Tooth mesh. Adapted from Henriot [43] and MAAG [19].

Table 4.2: Effects of the profile teeth modification. Results considering a load spectrum application.

	Don't considering profile modification			**Considering profile modification**		
Stage	$\mathbf{S_F}$	$\mathbf{S_H}$	$\mathbf{S_B}$	$\mathbf{S_F}$	$\mathbf{S_H}$	$\mathbf{S_B}$
1st	1.555	1.323	1.457	1.566	1.380	2.845
2nd	1.560	1.286	1.918	1.560	1.286	1.918
3rd	1.566	1.232	1.995	1.566	1.321	1.995

4.1.3 Power split gearbox's shafts assembly

This section is dedicated to the study and optimization of the transmission components assembly, of the second and third gear stages. After defining the transmission group layout and dimensioning those that shall be the ultimate geometric gears parameters, a somewhat careful issue shows up. It is concerns with the assembly of the gears with its own shaft.

Ultimately, gearboxes with multiple power paths and simultaneous meshing of several gears with the same pinion have been the object of study for too many authors (e.g. [44]), all of them deal with some geometrical aspects of the four simultaneous meshing conditions. Despite the use of multiple branch is widespread, there is no published information about the assembly — *one input-one output double branch* gearing — specific meshing type.

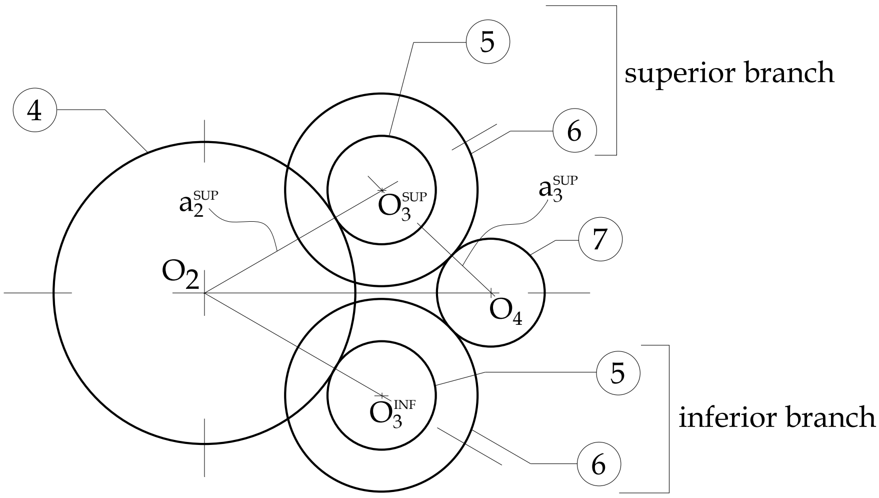


Figure 4.6: Second and third gearbox stages layout.

This follows an attempt to ensure a perfectly equal superior and inferior intermediate shafts. Notice that, both pinions, superior and inferior, are directly cut on its own shaft and a straight involute splines is applied for the transmission of motion to the subsequent gear, so a free angular motion between an intermediate gear and helical pinion is not allowed. Indeed, after having made a meticulously analysis, it has been found that there is a very particular position which allows the assembly of this type of gear arrangement; this position may or may not be symmetrical.

This problem would be more intuitive if all the gears of the different meshes had an even number of teeth; however, because of the cyclic loading this is not possible. It would lead to a limited number of teeth that could be operating throughout the machine life, not being the gear mechanical resistance used in full and accelerating the wear issues.

By analysing the Figure 4.7, it is understood that, given the numbers of teeth Z_1, Z_2, Z_3 and Z_4, it is necessary to make it possible to calculate the angular mismatch of the relative position of the gears, between the superior intermediate shaft and the inferior one.

It should be noted that it was considered the position in which the pitch point of whichever tooth of the gears 5^{SUP} and 6^{SUP} are coincident with the line segments connecting the centers O_2^{SUP} to O_3^{SUP} and O_3^{SUP} to O_4^{SUP} , respectively, as a *reference* position of the superior intermediate shaft. This can be seen in the Figure 4.7. It is also important refer that radial blue markers was used in order to demonstrate that the same relative position of the 5 and 6 gears was implemented, in both branches (superior and inferior).

In Figure 4.8, the angular mismatch of the 2nd and 3rd stages' gears, relative to the *reference* position, are θ_2 and θ_3 , respectively. They are the parameters that determined the position in which it is possible to assemble the ending shafts of the gearbox and for the calculation it is necessary to determine the *assembly angles*.

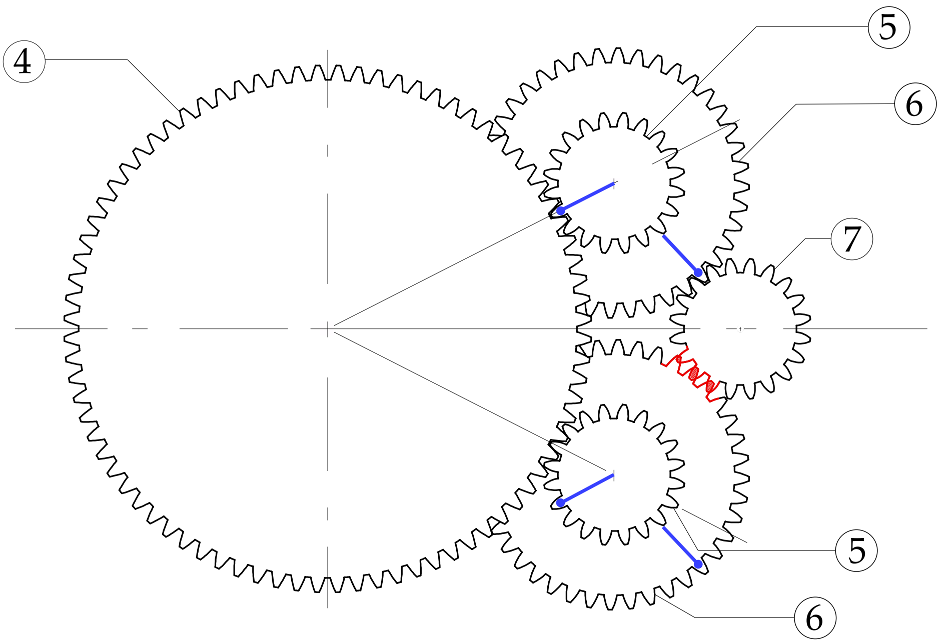


Figure 4.7: Reference positioning of seconf and third gear stages. Both intermediate pinion and gear are positioned exactly on the same orientation.

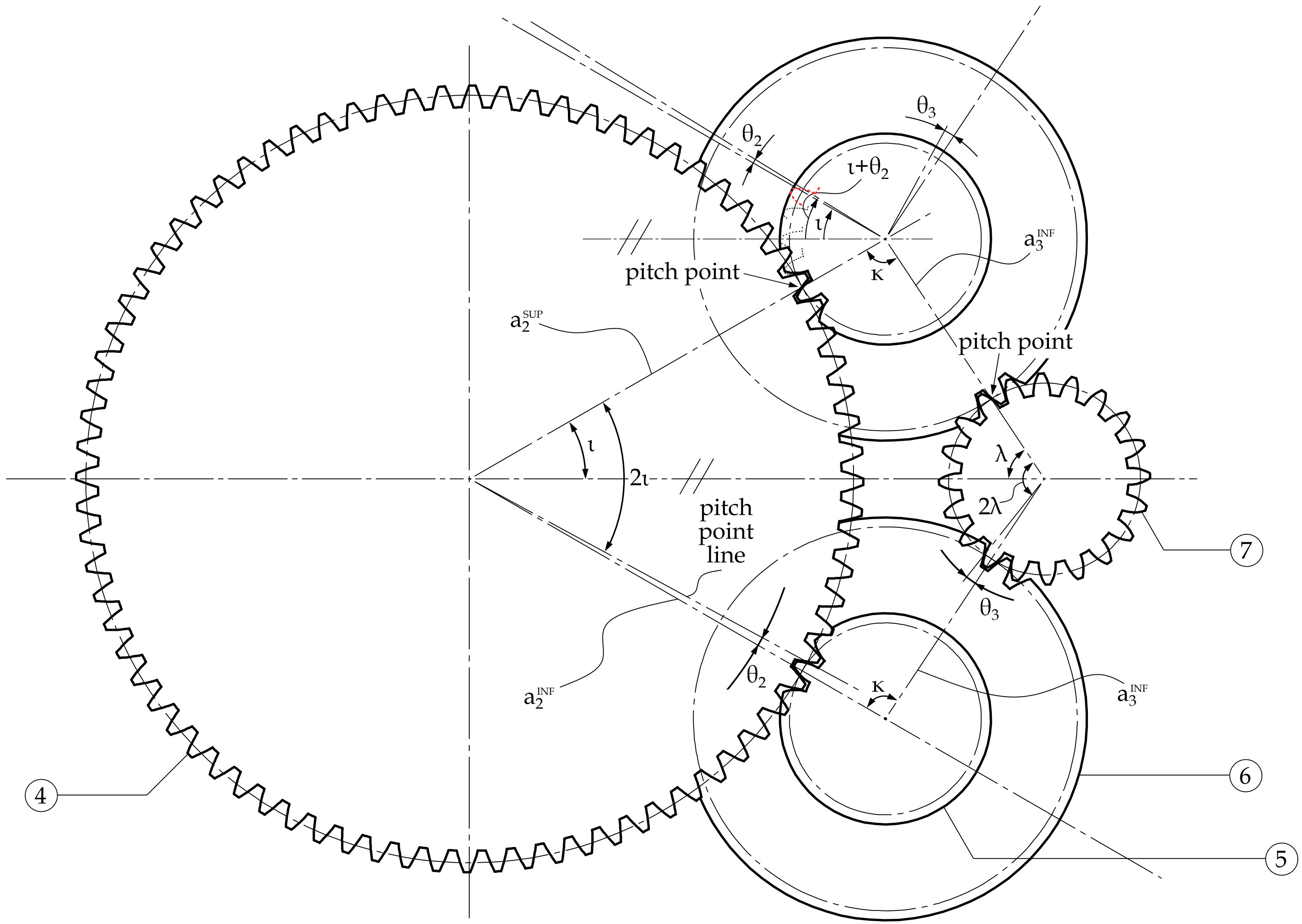


Figure 4.8: Second and third gearbox stages assembly parameters.

In order to simplify the following mathematical and graphical calculations, the angles ι, κ and λ will be defined as the *assembly angles.* They define the position of the several axis of the 2nd and 3rd gear stages (see Fig. 4.7), and the sum equals one revolution,

$$\iota + \kappa + \lambda = \pi, \tag{4.1}$$

where,

ι, is the angle between the center distance a_2^{SUP} and the horizontal plane containing the axes of the gears 4 and 7;
κ, is the center angle defined by a_2^{SUP} and a_3^{SUP};
λ, is the smaller angle between the center distance a_3^{SUP} and the horizontal plane containing the axes of the gears 4 and 7.

Since the gears of both meshes (superior and inferior) have the same geometrical parameters, e.g. *nominal module* m_n and number of teeth Z, they will also have the same size, making sure that the superior branch assembling angles are equal to the inferior branch idem. This results,

$$\iota^{SUP} = \iota^{INF} = \iota. \tag{4.2}$$

The same must be applied for the κ and λ angles.

Having presented the assembling angles, focus can shift to the geometrical gear conditions. That said, by dividing a revolution by the number of teeth of each gear, we shall begin to calculate the *angular pitch.* It is defined by the following relationship:

$$\gamma_n = \frac{2\pi}{Z_n}, \tag{4.3}$$

with γ_n being the *angular pitch* of any n gear (or pinion).

Now distinguishing the 2nd multiplication stage, the next step is to determine the number of teeth of the gear 4 comprised in the arc defined by the ι angle, $N_{2\iota}$, to which the following equation applies,

$$N_{2\iota} = \frac{2\iota}{\gamma_4}. \tag{4.4}$$

Its decimal fraction corresponds to the quantity, in number of teeth, of the angular displacement that pinion 5 must rotate to find the closest position that ensure meshing conditions. For example, for a number of teeth $N_{2\iota}$ of 10.25, pinion 5 should rotate 0.25 teeth in order to find the correct meshing position. Thus, this gear has to go through the arc corresponding to 0.25 teeth so that it meets the next tooth gap of the gear 4, avoiding the interference phenomenon described in Figure 4.7.

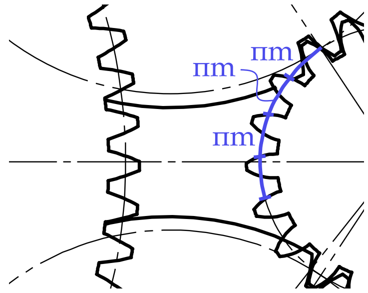


Figure 4.9: Curvilinear quadrilateral.

In Figure 4.9, the *curvilinear quadrilateral* [1] of the 2nd stage' meshing area under study is represented, which allows to understand more clearly the distribution of the circumferential distance of the mesh, typical in this type of gearing. Since pinion and gear of the same stage must have the same *nominal module* and the *circular pitch* is exclusively a function of the *nominal module*, then it will be the same for the pinion and the gear, too. This introduces the next step.

Considering the aforementioned and the scheme shown in figure 4.8, we conclude that the arc L_{2nd}, corresponding to the angular mismatch θ_{2nd}, is a portion of the *circular pitch* of the second gear stage and is defined as:

$$L = (N_{2\gamma} - \lfloor N_{2\gamma} \rfloor)\, p^w_{2nd}, \tag{4.5}$$

where p^w_{2nd} is the operating *circular pitch*.

We are now able to define the angular misalignment θ_{2nd}. It is the product of arc L, calculated in the previous equation, and the operation pitch radius of the gear 5, r^w_5,

$$\theta_{2nd} = L\ r^w_5. \tag{4.6}$$

Finally, remembering that the *reference* position of the pinion 5 is that in which, in the superior branch, the pitch point is coincident with the center distance a_2^{SUP}, in order to find the convenient position of the pinion 5, it must undergo a rotation of 2ι as a medium to move the active tooth from the lower to the upper zone of the axes. Thus, the pinion 5^{INF}, relatively to the position of the pinion 5^{SUP}, has a total misalignment, θ^{total}_{2nd}, of,

$$\theta^{total}_{2nd} = 2\iota + \theta_{2nd}. \tag{4.7}$$

Employing the same approach, the total misalignment of the gear 6 of the 3rd multiplication stage's inferior branch, θ^{total}_{3rd}, can also be calculated. It presents the following definition,

$$\theta^{total}_{3rd} = 2\lambda + \theta_{3rd}. \tag{4.8}$$

All of this culminates in the discovery of the global angular mismatch Θ.

Note that, considering the clockwise direction as positive, the global angular mismatch Θ will be no more than the sum of total misalignments θ^{total}_{2nd} and θ^{total}_{3rd},

[1] definition in accordance with [44]

$$\Theta = \theta_{2nd}^{total} + \theta_{3rd}^{total}. \tag{4.9}$$

in which for both of them, the plus sign indicates a clockwise rotation and vice versa.

4.2 Shafts

Transmission shafts, or typically just *shafts*, are used virtually in every piece of rotating machinery to transmit rotary motion and torque from location to another, [45]. This is, just like in this project, shafts typically carry gears transmitting the rotary motion via meshing gear and consequently multiple stresses are led to the shaft. The loading on rotating transmission shafts is principally one of two types: torsion due to the transmitted torque or bending from transverse loads at gears.

These two loads often act in combination. They can be steady (constant) or may vary with time. In the context of this work, this gearbox is faced with a situation of steady torsion stress and alternating bending ones. So, considering an infinitesimal element of the shaft, it will always be affected of the same torque in any position, during the rotation characteristic of the mechanism operation. In contrast, although the steady transverse load, the same stress element on the shaft surface will be affected by a fully reversed bending stress. This happens because the forces associated with the contact of the teeth, during the meshing gear, cause a transverse deformation in the shaft which, due to its rotational motion, will vary between a maximum and a minimum value. So, the infinitesimal elements go from tension to compression each cycle as the shaft turns. It should also be noted that due to the use of helical teeth a constant axial force is also transmitted to the shaft. This combined effect of stresses points us to the problem to which this section is devoted: the fatigue failure.

Fatigue is defined as a phenomenon of progressive damage of materials subject to cyclic variation of stress or deformation. The study of the phenomenon is of crucial importance in the design of machines and structures, since the great majority of the failures observed, during operation, involve the fatigue effect. Among the components subjected to repeated loading and unloading, the gearbox' shafts stand out. They are under a alternating bending stresses caused by the cyclical motion characteristic of their rotation. Although there is an analysis of the static load, in this type of cyclical motion it is insufficient because materials tend to have a lower resistance when under repeated loading and unloading.

This issue follows the optimization of the shaft so that it can correctly house and locate components such as rolling bearings. In other words, in the previous chapter, the shafts considered were just metal cylinders in order to simplify the comparison. However, real gearbox shafts are not that simple, it commonly require features such steps or shoulders were the diameter changes to accommodate attached elements such as gears and rolling bearings. That features are necessary to provide accurate and consistent axial location of the attached elements avoiding them to slide along the shaft for example. Other general features are the keyslots for key joints or even retaining rings. Although the fatigue mechanism has different successive phases, its beginning usually occurs in the surface, since in general it is where the effect of the eventual stress concentrations,which are result of the previous discontinuities, is maximum. It should be the main objective to reduce these surface stress concentrations without correct assembly and operation being changed or suppressed.

There are several ways to achieve this, one of them is, in the sections where the shaft diameter varies, consider a gradual decrease of the dimension through the match radius usage; however, this solution is sometimes inadequate. It should be noted that for mating parts that do not have relatively large counterbore, the application of this mechanism can hinder total and correct contact between the components. Because of this, circumferential cuts were used with different shapes of sections, adapted to each of the situations. In such a particular cases, relief groove was applied. It can broadly be defined as a clearance groove of specified form and dimensions created by removing material at an inner corner of a rotationally symmetric workpiece.

Following [46], the FKM guideline for analytical strength assessment of mechanical components has gained an increasing interest for the last decade in the industry, because it describes a general procedure directly applicable in an industrial design office. It allows in particular a sound interpretation of finite element analysis, making an efficient bridge between stress results and a trustworthy safety margin. For that reason, it was the method considered for the shaft calculation.

4.2.1 Assessment of the fatigue strength using nominal stresses: general safety factors

Starting with the fatigue dimensioning coefficients, the FKM standard was followed [47]. According to this, in general, the safety factor consists of partial safety factors with regard to the load and to the material,

$$j = j_S + j_F, \tag{4.10}$$

where, j is the safety factor; j_S is the load factor; and j_F is the material factor.

It considered that safety factors are valid on the condition that the characteristics strength values exhibit an average probability of survival of 97.5 percent. And, in these conditions, the j_S factor applied can be,

$$j_S = 1.0, \tag{4.11}$$

which was the one considered.

For non-welded components, the material safety factor for fatigue strength depends on the possibilities of inspections and the consequences of failure is presented in the following table.

After that, and considering the previous table, a **1.35** j_F factor was chosen.

4.2.2 Assessment of the fatigue strength using nominal stresses: individual safety factors

The safety factors used for the assessment of the static strength have different values, depending on the probability of the occurrence of the highest stress or the most unfavorable

Table 4.3: Material safety factors j_F for non-welded steel, [47].

j_F		**Consequences of failure***		
		severe	mean	moderate
Regular	no	1.50	1.40	1.30
inspections *	yes	1.35	1.25	1.2

* more detailed information should be verified in FKM standard.

stress combination and depending on the consequences of failure. The basic safety factors applied to ductile materials are presented in Table 4.4.

Table 4.4: Basic safety factors for ductile materials. Ductile macterials are definine by FKM as a material with elongation at break at least of 6 percent, [47].

j_m j_p j_{mt} j_{pt}		Consequences of failure		
		high	mean	moderate*
	high	2.0	1.85	1.75
		1.5	1.4	1.3
Probability of		1.5	1.4	1.3
occurrence of		1.0	1.0	1.0
stress or stress	low	1.8	1.7	1.6
combination	***	1.35	1.25	1.2
**		1.35	1.25	1.2
		1.0	1.0	1.0

* Moderate consequences of failure of a less important component in the sense of “no catastrophic effects” being associated with a failure.
** Generally with reference to the magnitude of the load, not the frequency.
*** Including exactly estimable loads which can safely be assumed to occur infrequently due to e.g. the testing and assembling conditions.

Due to the high magnitude of the WT load, it was considered a high probability of occurrence of stress and a mean consequences of failure. Bearing that in mind, the following basic safety factors was set (tab. 4.5).

Table 4.5: Basic safety factors for the final gearbox solution.

j_m	j_p	j_{mt}	j_{pt}
1.85	1.4	1.4	1.0

4.2.3 Shaft Results

Once the shaft diameters were modified to accommodate all of the components and after these stress concentration relief solutions were applied, the results of the shaft calculations are shown in the table 4.7.

Table 4.6: Pinion shafts material properties: sun shaft, Intermediate shaft (inferior and superior) and high speed shaft, [33].

Parameter	Unit	Value
DIN material	**-**	18 CrNiMo 7-6
Heat treatment	**-**	Case-carburized
Yield strength	**MPa**	≥ 735
Tensile strength	**MPa**	≥ 885

Table 4.7: Shaft deflection and safeties, considering load spectrum. The deflection criteria followed is suggested by Movnin, [48].

Shaft	**Max. deflection*** μm	δ/m_n	δ/l	**Min. static safety*** **-**	**Min. fatigue safety*** **-**
Planet shaft	280	0.017	-	24.73	28.72
Intermediate (sup.)	420	0.084	0.0006	4.0	1.70
Intermediate (inf.)	285	0.057	0.0004	5.45	2.25
High speed	380	0.076	0.0005	3.27	2.13

* Results for the highest load factor bin. Relief grooves in accordance with: *ISO/FDIS 18388 - Relief grooves: Types and dimensioning.*

4.3 Rolling bearings

In most cases, friction and the resulting wear are undesirable. So, allowing relative rotary motions, the aim of rolling bearings is to reduce friction through suitable material combinations for components and the use of lubricants. They can play the roles of support, guidance or even both. In this optimization process, various aspects—such as bearings' dimension, function, arrangement, interference fits, group of clearance and cleanliness — are discussed. KISSsoft® database proposes various rolling bearing from many manufacturers; however, it makes little sense to order bearing from different sources and, as such, SKF® was chosen to be the sole supplier of bearings for this application.

By the particularity of its use, the first step is to discuss the planet carrier's bearing arrangement. Because of the balanced configuration characteristic of planetary gear, the planet carrier is under forces mainly resultants of its own weight and do not result from the internal forces of the multiple meshes. Having said that, these rolling bearing has guidance as a key location function, i.e. they principally comprise the function to accurate an appropriate position of the planet carrier, moving relative to the sun and ring shaft |[2]. Although low (relative to the overall gearbox's forces), their most important load (due to its own weight) is the bending moment. So, a *cross locating* |[3], with tapered roller bearing,

[2] as ring shaft we understood the annular body in which the internal teeth of the ring gear was milled.
[3] According to the 61400 standard of the [34], in cross locating arrangements the shaft is axially located in one direction by one bearing position and in the opposite direction by the other bearing position.

bearing arrangement was considered. When a load acts eccentrically on a bearing, a tilting moment occurs and, according to bearing technology providers — [49],[50] and [51] — , a paired single row tapered roller bearings can accommodate that kind of moment loads, and that is why they was selected.

In tapered roller bearings, a radial force always generates an axial force. Thus, they must be mounted in pairs, with a symmetrically arrangement, in order to balancing the forces. **X** and **O** arrangements [4] would be considered as a possible symmetric bearing arrangement, which the vertex of the cone formed by the normal lines to the tapered roller axis point inwards and outwards, respectively.

It is known that the greater the distance between the points of convergence of load lines (fig. 4.10), the wider effective support of tilting moments these rolling bearings have. For this reason, and considering the substantial distance between both of planet carrier's rolling bearing, a **X** arrangement was considered, in preference of the another one mentioned. The condition of motion legitimizes this option, too.

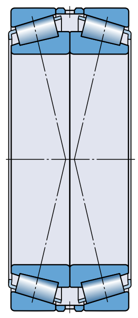
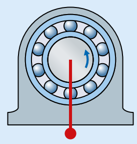
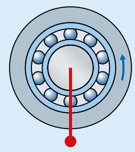
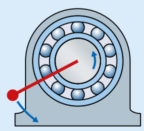
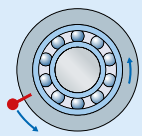


Figure 4.10: Face-to-face tapered roller bearing arrangement and its normal lines. Adapted from: [49].

The bearing condition of motion can be classified as *Point load* or *Circumferential load*, depending if the rolling bearing ring remains stationary relative to the load direction or not. In other words, if the ring (inner or outer) describes a condition in which it rotates relative to the load, every point over the raceway is subjected to load over the course of one revolution of the bearing. In this case, a *Circumferential load* is said and a tight fit should be provided for the ring in question. Table 4.8 shows several condition of motion configuration.

[4] also called *face-to-face* and *back-to-back* bearing arrangement, in accordance to [49].

Table 4.8: Conditions of rotation and loading by SKF®, [49].

Operating conditions	Schematic illustration	Load condition	Example	Recommended fits
Rotating inner ring Stationary outer ring Constant load direction		Rotating load on the inner ring Stationary load on the outer ring	Belt driven shafts	Interference fit for the inner ring Loose fit for the outer ring possible
Stationary inner ring Rotating outer ring Constant load direction		Stationary load on the inner ring Rotating load on the outer ring	Conveyor idlers Car wheel hub bearings	Loose fit for the inner ring possible Interference fit for the outer ring
Rotating inner ring Stationary outer ring Load rotates with the inner ring		Stationary load on the inner ring Rotating load on the outer ring	Vibratory applications Vibrating screens or motors	Interference fit for the outer ring Loose fit for the inner ring possible
Stationary inner ring Rotating outer ring Load rotates with the outer ring		Rotating load on the inner ring Stationary load on the outer ring	Gyratory crusher (Merry-go-round drivers)	Interference fit for the inner ring Loose fit for the outer ring possible

Following Table 4.8, it can be seen that for the planet carrier, with stationary outer ring and load direction rotating with the inner ones, as far as possible there should be a tight and loose fit in the inner and outer rings, respectively. For that, once again, the ***face-to-face*** is the better option. Notice that, with the tight fit in inner rings, we have freedom to adjust both outer rings to apply the needed preload. SKF Group® catalog [49] says one way to preload the bearing arrangement is applying a negative operating clearance, that is accomplished with the employment of a laminated shim.

An equivalent analysis must be done for the planet gear shaft.

Conversely what happens in the planet carrier, in the planet gear shaft a non-variable load direction in relation to the inner ring takes place. As such, the *circumferential load* is on the outer ring; and, therefore a **back-to-back** arrangement must have been applied. It is important to note that in the case of the planets, although they cancel out axial forces at the two points of contact of them with the sun and the ring, respectively, these forces create a tilting moment and that is why tapered roller bearings were used. The **O** arrangement also ensures greater tilting moment stability, since the points of convergence of load lines are further apart than they would be in an **X** arrangement.

Regarding the sun shaft, because of the single helical planetary gear, it has high combined axial and radial loads; therefore, a combination with a radial and an axial forces carrying rolling bearing has been selected. Assuming a *locating and non-locating* bearing arrangement for this shaft, since the locating function is already being performed by the match of the radial and axial bearings, a non-locating bearing has to be found. It shall provides radial support only, allowing axial displacements to accommodate shaft length thermal variations. The one selected was the spherical roller bearings, this in addition to allowing adaptability to the effects of thermal expansion, it also allows angular self-alignment which provides some freedom of flotation in the sun, adjusting this to the multiple teeth meshing with the different planet gears. This ensures a uniform distribution of power.

According to the Table 4.8, for a stationary outer ring a *circumferential load* on the inner ones is verified; then, a tight fit was applied on both inner rings of the cylindrical roller bearings and the axial ones.

For the intermediate bearings, special SKF Explorer bearing was selected. They are designed for applications such as gearboxes in wind turbines and others, [49]. It is known that bearings with a cage can accommodate heavy radial loads, and the maximum example of this are the full complement bearings. SKF® high-capacity cylindrical roller bearings combine the high load carrying capacity of a full complement bearing with the high speed capability of a bearing with a cage. Its window-type metal cage is designed so that its cage bars are displaced relative to the roller pitch diameter, enabling the rollers to be placed closer to each other, creating room for addition all rollers. A **NCF .. ECJB** design was chosen. With an inner ring centred cage, they are used to locate the shaft axially in one direction and eventually to accommodate axial displacement of the shaft relative to the housing in the opposite direction; so, a symmetrically configuration must be applied in order to accommodate shaft length thermal variations. Don't should be forgotten that a two cylindrical roller bearing can be considered, due to the cancel out of the axial load by the double helical gears.

Finally, for the high speed shaft, some important details had to be analysed and optimized. With a relatively restricted outside diameter, because of the available gear center distance, a solution with a high radial load capacity (effect of the double branch with double helical gear) had to be found. It should be noted that, since the high speed gear is not positioned at half of the shaftway, the rotor and generator side bearings will be under different conditions of load, being last the most critical one. Although the tapered roller bearings not being the first choice for the higher speed shaft of a gearbox, the output shaft of this WTG is supported at the two ends that are significantly spaced apart from the point of load application (the output pinion). This causes significant bending effects in the supports, which are not compatible with the use of cylindrical roller bearings or even deep groove ball bearings.

Thus, after testing several possibilities, the chosen configuration was: a cylindrical roller bearing and a paired single row tapered roller bearing on the rotor and on the generator side, respectively. This solution allows to distribute the load through the two bearings, without asking for a bigger rolling bearing.

Regarding bearing life calculations, instead of using the manufacturers formulas, as in the previous chapter, a more accurate method was used, specified in ISO/TS 16281:2008 and ISO 281:2007, which calculates bearing life considering the influence of several factors such as lubricant contamination, filtration, shaft tilting and misalignment. Using this method, two new input parameters must be given: the filtration ration $\beta_{x(c)}$, with particle size x in $\mu m(c)4$ according to ISO 16889, is the most influencing factor when selecting diagrams and equations for circulating oil systems with on-line filters; and, the contamination level, which corresponds mainly to the condition of the oil before it passes the on-line filter. Recommended values from IEC 61400-4 [34], were use to set this parameters. According to them the steady state cleanliness level for a gearbox in constant operation shall not be worse than -/17/14, with a typical value of the beta ratio greater than or equal to 200.

It should be noted that tighter cleanliness values can be considered with the consequence of increasing costs. Nevertheless, the greater cleanliness values, the shorter the lifetimes for the bearings are expected, so we are working on the safety side.

Table 4.9: Bearing service lifetime and safety levels. Results considering the oil cleanliness level ISO/TS 16281 [52] and the load spectrum.

Shaft	**Min. static bearing safety**	**Min. bearing service life**	**Max. pressure**
	-	x 1000 *hours*	MPa
Planet shaft	3.33	198	2 285
Intermediate (sup.)	4.67	476	2 124
Intermediate (inf.)	6.34	> 1000	1 888
High speed	9.60	> 1000	1 636

Table 4.10: Rolling bearings properties and dimensions.

Shaft	Position	Designation	Dimension series	d	D	B	Mass
				mm	mm	mm	kg
High speed	Rotor side	Cylindrical roller bearing (single row)	SKF NU 320 ECJ	100	215	47	7.47
	Generator side	Tapered roller bearing (single row)	SKF 30320 J2	100	215	51.5	7.96
Intermediate shaft	Double	Cylindrical roller bearing (high-capacity)	SKF NCF 2328 ECJB	100	300	102	35.5
Planets	Double	Tapered roller bearing (single row)	SKF 32248 J3	240	440	127	81.5
Sun	Rotor side	Spherical roller bearing	SKF 23056 CCK/W33	280	420	106	52.5
	Generator side 1	Cylindrical roller bearing (high-capacity)	SKF NCF 2244 ECJB/PEX	220	400	108	58
	Generator side 2	Cylindrical roller thrust bearings	SKF 89434 M	170	340	103	51.9
Planet Carrier	Double	Tapered roller bearings (single row)	SKF EE 843220 /843290	558.8	736.6	88.108	92.5

Although the shaft calculation method proposed by DIN 743 was not mainly followed, all the shafts were checked according to it. It was verified that the fatigue safety was always the critical factor presenting the lowest value of 1.75 for the superior intermediate shaft. This value is within the normalized range suggest by the IEC 61400-4, [34].

4.4 Spline joints

Increasing demands for high speed and compactness in new machines generally increases the power employed in their systems and, consequently, the noise. That result in a focused attention on the problem of its control. Defined as *"any unwanted sound"* by [53], the noise is closely linked with the vibrations set up within the solid structure. It is essential to differentiate clearly between vibrations and noise actually generated in air. Here, noise refers only to airborne sound, but its reduction may involve some operations on solid objects. A good example of this is the use of straight cylindrical involute splined shafts. This is a smoother way of motion transmission than a key.

Splines are used to couple two coaxial components, transferring power between them. Following the ISO definition, a spline joint is presented in terms of, *connecting, coaxial elements that transmit torque through the simultaneous engagement of equally spaced teeth* *situated around the periphery of a cylindrical external member with similar spaced mating spaces situated around the inner surface of the related cylindrical internal member* [54].

In the context of this thesis, they have the function of replacing the action of a key in the transmission of torque from the shafts to the multiple gears. According to [55], the use of splines is the best solution whenever large amounts of torque to be transmitted. Although splines resemble gear teeth and are cut with the equivalent tools, their action is somewhat different, in terms that, (with spline) the teeth all fit together.

This leads us to choose the type of interference fit for the spline teeth. Three types of fits ate illustrated in Figure 4.11. In flank-centred connections, the flanks of the teeth serve to transmit the forces as well as to centre the parts, and that is why this type of fit was chosen. In fact, in this application a large amount of power is needed to transmit, using the flanks area to its transferring, a more active area will be consider and a less concentration of load shall be applied. Also, with the diameter-centred fitted splined connections, a over-determination of the centering needs to be prevented with the chose of enough backlash. A loose choice of it could compromise the correct transmission of motion, since the contact between the shaft and the hub was not verified.

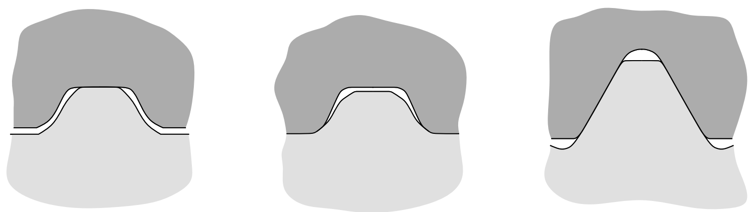


Figure 4.11: Example of spline fits, [55].

The ISO 4156 standard [54] list three standards for involute splines, depending upon the nominal pressure angle employed are 30° ,37.5°|[5] and 40°. The 30 ° pressure was selected, it is most often used when the two members are allowed to slide. Note that due to the use of the double helix it is necessary the gear to be able to slid freely along the shaft so that it can self-center with its own pinion, balancing the load. It can be also referred that higher pressure angles normally use large number of teeth, which makes the cutting process time-consuming.

The external and internal diameters always have clearance and do not contact each other, Fig. 4.12, i. e. the diameters of the tip and root circles of the shaft differ from the respective diameters of the hub and the exact dimensions can be consulted in the second part of the ISO 4156 standard, [56].

[5] For electronic data processing purposes, the form of expression 37.5° has been adopted instead of 37°30’

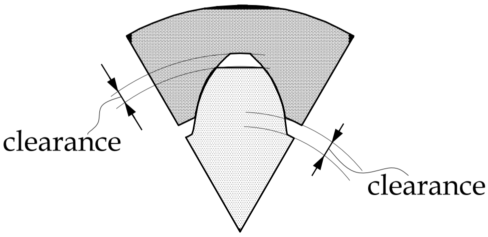


Figure 4.12: Radial clearance of a flank fitted spline joint, [54].

Contrary to what happens with the gear, with splines there is no rolling action and that means wear is no problem in their design, and bending stress idem. According to [55], the most frequent type of failure is torsional failure of the shaft. However, a stress and strength analysis was applied in order to quantify the safety coefficient present in the use of splined shafts. For the dimensioning of these components, the spline connection module in KISSsoft® was used. The calculation method used is the ISO 4156:2005.

Most of the input parameters are imposed by previous choices such as material and diameters of shafts, gear facewidth, and even by specific parameters such as the diameter of the cutting tool and, consequently, the exit length for it. Consulting some tools catalogues, a milling tool diameter of 80mm was found, for a 3mm and 5mm nominal module of the involute spline, fig. 4.13. With this, it was possible to estimate the overall length of the spline, adding the exit length required for the cutting tool to the gears facewidth. It should be noted that this exit length is only for drawing purposes, for the purpose of stress analysis only the gear facewidth was considered, which is the real active transmission area.

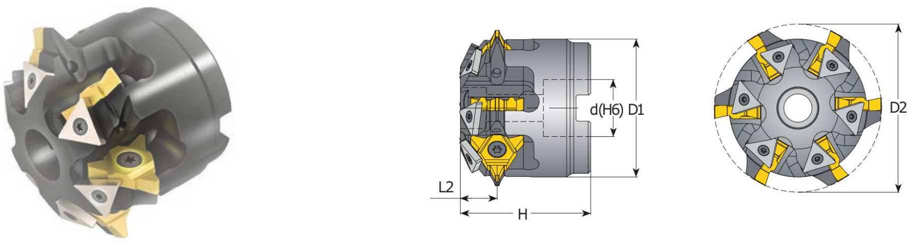


Figure 4.13: Gear milling toll. Image for illustration only. Final D1 and D2 values will be supplied with the toolholder. D2=80, [57].

Table 4.11: Desgination of spline joints dimensioned in KISSsoft tool.

Spline joint	**Designation**
Sun shaft-Low speed gear	ISO 4156 34Z x 8m x 30R
Intermediate shaft- Interm. gear	ISO 4156 61Z x 3m x 30R

4.5 Housing

The housing's role is to form a strong base in which it is possible to mount the bearings, support the gears and shafts, and create an environment and space where a satisfactory lubricant may be introduced to lubricate and cool the gears. It also have the function of to house several other components such as accessories and parts near or common to the gearing.

The housing will be manufactured in a continuous casting process, so the design driver was the use of round shapes. When round shapes weren't possible to apply, a tilt angle between the mentioned surface and the exit direction of the part was considered. This essentially occurs in main (input and output) lids, fig. 4.14.

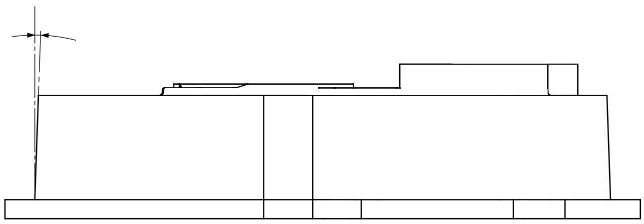


Figure 4.14: Tilting angle of main lid (generator side) surface, [57].

It will be seen, in a general race, the housing is formed by:

- a main body;
- a main input lid;
- a main outlet lid;
- five inspection covers;
- four secondary external covers;
- three secondary internal covers;

The main body was thought to be robust so that it can to prevent excessive distortions resulting from thermal variations, mechanical loading, or both. It has a pivotal wall separating the two parts of the gearbox — the planetary stage and the remainder two with power split. For the correct operation of the mechanism it was not needed to divide the two parts of the gearbox, however due to a matter of compactness it was necessary to create several supporting points for different rolling bearings, namely matched tapered roller bearings of the sun shaft and cylindrical roller bearing of the intermediate and high speed shafts, which are positioned on the rotor side. It should be noted that the main body is also responsible for the transition from the circular section of the ring gear rim to the second and third stages' double branch configuration.

The main lids, input and output, have the function of providing split planes for gear mounting and dismounting. The alignment of each lid with the main body is allowed by the application of tapered dowel pins and it shall be employed a flat metal-to-metal joint maintained oil tight with suitable sealing compound. Both lids have multiple removable inspection covers for field inspection of the full facewidth of all gear meshes. By recommendation of [15], the tapped holes for the bolts of the inspection covers should be blind. Because of the configuration of the planetary stage it may be difficult to inspect the gears and, for that, borescopes using fiber optics can be used to inspect them. Particular attention should be paid to the fact that the input lid contains supporting torque arm to prevent the housing from rotating. The configuration chosen for them design was an attempt of robustness and versatility, in that several points can be used for the gearbox attachment. However, there is an entire analysis of its dynamic behavior that must be done, allowing a complete study of the stiffness' gearbox foundation. This was not possible to perform as it would require a substantial amount of time for small improvements.

The secondary covers are all used for the fixing of rolling bearings and to provide lubrication paths for their raceways. The external covers are coupled on the output lid. It should be also employed suitable sealing compound (like in the split plane of the main body) in the joint plane of the external secondary covers with the output lid.

For the last, the housing have internal and external lines of oil circulation, even the walls place by on its own paths for the lubricating system. This will be carefully explain in ensuing sections.

Following [15], all gears should be totally enclosed in a cast ductile iron or cast steel housing. So, it was selected the nodular cast iron GGG40, Table 4.12 .

Table 4.12: GGG40 mechanical properties, [33].

Yield strength MPa	**Tensile strength** Mpa
≥ 245	≥ 392

4.6 Other components

4.6.1 Shaft seals

Retaining lubricant and preventing any contaminants from entering into a controlled environment are the most general roles of the seals. Indeed, the propagation of particles inside the gearbox would be a decisive factor for the operation of the components such as rolling bearings and gears. In addition of providing surface wear, their storage can creat sticky and adhesive sediments. The seals applied in this works were fully selected from [58].

The chosen seal designs are the **BAUMX7** and **Radiamatic R35** for the input and output shafts, respectively.

The **BAUMX7**, with a improved chemical resistance and thermal stability, is a reliable seal even with increased thermal expansion. With a preferred usage for synthetic oils temperature higher than 80 °C, it is a good choice for an oil injection of 65 °C. Notice that, for this inlet temperature it is expected a higher return temperature. The **BAUMX7** seals include a steel spring acting as a pretensioning element for the seal lip. The spring provides a defined uniformly distributed contact, guaranteing a continuously sealing. A *only one lip* design allows a low friction solution, usable for circumferential speeds of 10 m/s or more which is approximately the linear speed of the output shaft.

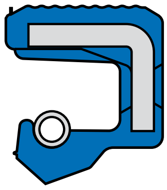


Figure 4.15: **Radiamatic R35** shaft seal.

Although according to the chart provided by the manufacturer, presented in Figure 4.16, the speed and diameter of the shaft suggest the use of NBR material, **FKM** material was chosen. Since the seals are a low cost component, the price difference does not justify the choice of a less noble material. With this decision, we are always going to extend the safety of the mechanism.

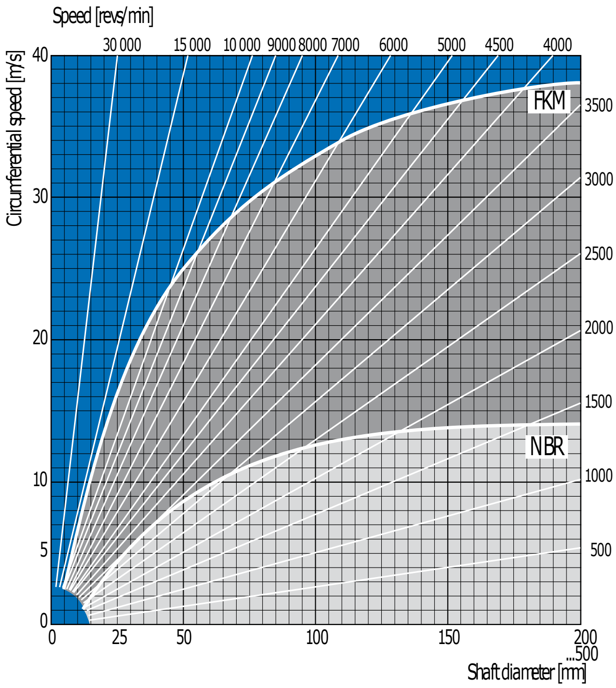


Figure 4.16: Circumferential speed allowed by NBR and FKM SIMRIT® shaft seal materials.

According to [59], shafts seals rings with additional functions like **Radiamatic R 55** design with a deflector lip, are used to seal the main bearings in wind turbines. Consulting the several catalogues we have found that there are no far information about its geometric and mechanical specifications. So, a **Radiamatic R 35** in combination with a standard **V-ring** was chosen, Fig. 4.17. According to the same author, it is a often used solution in this kind of application by substitution of **R 55**.

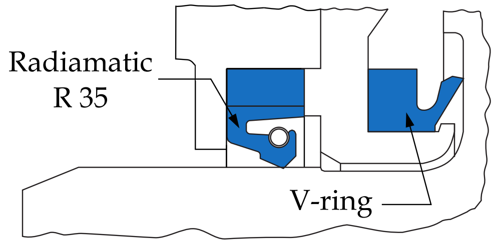


Figure 4.17: Shaft seal arrangement. Radiamatic R 35 in combination with a standard V-ring.

With this configuration it can roughly be said that the **R 35** seal will be essentially responsible for protection against external contaminants and the **V-ring** will be responsible for the interaction with the lubricant, inside the gearbox. Thus, the most noble material (FKM) was left for the **V-ring**, whereas **Radiamatic R 35** seal will be made on the NBR material. It should be noted that with this sealing material choice we have greater dimensional freedom, as for this magnitude of size less fluoro rubber (FKM) solutions than nitrile rubber (NBR) ones are available.

Given the low input circumferential speed there will not be the problem of high friction or high heat generation in the contact between the components, so a solution with two points of contact between the seals and the rotating shaft is applicable.

Table 4.13: Chosen SIMRIT® seals characteristics, [58; 60; 61].

Manufacture	Simrit	Simrit	SKF
Model	BAUMX7	Radiamatic R 35	VL
Size, mm	100x130x12	500x544x20	510x523x10.5
Material	Fluoro rubber	Nitrile rubber	Fluoro rubber
Sealing configuration	Straight	Straight combined with V-ring	
Surface speed shaft, m/s	≥ 10	≤ 20	≤ 6.5
Operating temperature, °C	-25 to 160	-30 to 100	-20 to 150

4.6.2 Retaining rings

A Retaining ring (or circlip) is a fastener that allows the fixture of components to prevent them from sliding freely along the shafts and housing. Although not always possible due to the high magnitude of size of this gearbox, in this application some retaining rings were used to fix some smallest bearings. The rings' dimension and mounting were determined following DIN 471 and DIN 472 for shaft and hub rings respectively. They were selected from the online catalog of Seeger-Orbis®, [62].

It will require a close look at one of the high speed bearings positioning, namely the generator side one. Explaining better, according to DIN 472, the external diameter of 215mm is not a standard bore diameter. So, it forced the use of a 220 retainer ring with an abutment collar inserted between it and the rolling bearing external ring in order to not subjected the retaining ring to excessive transitory bending moments, Fig. 4.18.

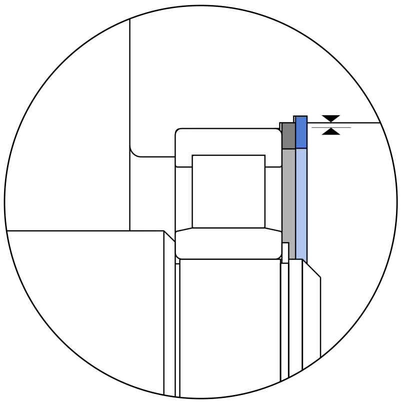


Figure 4.18: Retaining ring and spacer washer arrangement.

4.6.3 Spacer sleeves

In this work, two types of spacer sleeves must be essentially distinguished: the *purely positioning spacer sleeves* and the *lubricating spacer sleeves*. Them use for the positioning of adjacent components enables faster assembly and disassembly and simplifies the machining of shafts and housing bores. Actually, both have the function of maintaining the distance between the components mounted on the shafts and, combined with retaining rings, fix their position; however, in addition to allowing axial positioning, the *lubricating spacer sleeves* have the particular characteristic of to accommodate lubrication paths for the supply of matched rolling bearings, namely tapered roller bearings.

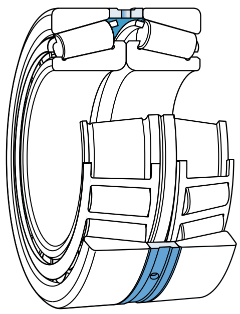


Figure 4.19: *Oil distribution* spacer sleeve of a paired single row tapered roller bearing, [49].

Since the *purely positioning spacer sleeves* are not loaded, a regular C 45 E (Ck 45) carbon steel has been chosen for these components. The same is not applied for the *lubricating spacer sleeves*, which due to the thermal loads and the interaction with the lubricant calls for special attention.

Table 4.14: EN C 45 E (DIN Ck 45) untreated mechanical properties, [33].

Yield strength MPa	**Tensile strength** Mpa
≥ 325	≥ 580

The lubricants generally used for rolling bearings do not have a detrimental effect on spacer sleeves properties. Yet, with some synthetic oil base and lubricants containing EP additives, the mechanical integrity of certain materials can prove inadequate. Consequently, a non-alloyed structural E555 (St52.0) was chosen. It can be used at operating temperatures up to 300 °C and it isn't affected by the synthetic oil-based lubricants, or by the organic solvents used to clean bearings, [33].

Table 4.15: EN E355 (DIN St52.0) mechanical properties, [33].

Yield strength MPa	**Tensile strength** Mpa
$\geq$ 335	$\geq$ 490

At last, in order to be sure that the bearings are not “clamped”, adequate axial internal clearance has been performed, between the *lubricating spacer sleeve* and its adjacent rolling bearing races. That is, when adjusting tapered roller bearings against each other it is necessary to ensure that the rollers are able to rotate freely at their correct position and an extra load is not being imposed by an oversized width of the spacer sleeves.

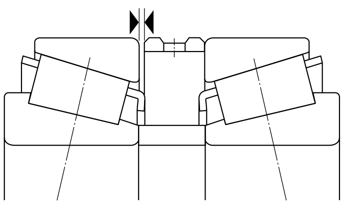


Figure 4.20: Internal axial clearance of a paired single row tapered roller bearing, [49].

Part III

Chapter 5

Lubrication system

5.1 Introduction

Our world, as we know, strongly depends on mechanical devices that are closely related with lubrication issues. By other words, whether in our homes or at work, all of us are surrounded and dependent on the mechanism that, for its proper functioning, requires lubricant films for both friction and heat dissipation issues. Like this, lubrication and the knowledge of lubricants no only are a subject of interest for us but they are also critical to the cost effective operation and reliability of machinery that is part of our daily lives. This section builds upon the lubrication fundamentals that was followed for the lubricating system design and dimensioning.

Knowing that the name *petroleum* is derived from the latin petra (rock) and oleum (oil), it is easy to conclude that it present in among rocks in the earth. keeping pace with the alwasys more demanding technology available, petroleum has been worked and combined in various ways taking various forms and applications today. Lubricating oils for wind turbine gearbox is one of them. As stated in the previous chapter, the lubricant chosen for this project was the synthetic oil **Castrol Optigear Synthetic X 320**. Considerable attention has been focused on synthetic lubricants since the introduction into the retail market of synthetic-based engine oils and its application extends back over many years. Interests is synthetic lubricants were due to their ability to resist burning to a greater degree than mineral oils and to provide equipment protection advantages under extremes of operating conditions, [37]. In the context of this work, its advantage includes extended service, low temperature starting, high properties stability with temperature and broad temperature application range.

Considering this application, the elements of machines that require lubrication are mainly gears and rolling bearings. These elements have fitted or formed surfaces that move with respect to each other by sliding, rolling, and advancing and receding, or by combinations of these motions; thus, these elements must be lubricated to prevent or reduce the actual contact between surfaces which may cause high frictional forces leading to high temperatures, and possibly wear or failure. This is particularly important in this work since the expected lifetime of an wind turbine is about 20 years and, consequently, the same is expected of the gearbox. It should never be forgotten that in machines of this size the stopping times are considerable and therefore the cost of interrupting normal operation for maintenance or replacement of components is very high. The implication of many stops increases the market price of the gearbox, which can lead to a lack of interest about the same.

Lubrication films may be classified in three main different kinds — fluid, thin or solid — but in this work it was only considered a *fluid* type of. With *fluid films* we have the enough thickness to ensure completely separation of the load carrying surfaces during the operation. Despite of the drag effect of the lubricant that sometimes occurs, here it is a practical minimum as well as the wear theoretically does not occur because there is no mechanical contact. Since a convergent zone exists immediately before these areas of contact, a lubricant will be drawn into a contact area and can form a hydrodynamic film, more precisely, an elastohydrodynamic film. That is typical of the contact between two different gears that elastic deformation of the material occurs before the film can be formed. As such, this type of lubrication mode is a particular case of hydrodynamic lubrication being awarded with the prefix "elasto". It is common to use only the acronym EHD|[1].

Gears have the function of transmitting power from one rotating shaft to another along the entire kinematic chain, from the input of the gearbox to its output. With respect to lubrication and the formation and maintenance of lubricating film, the gears can be classified into several types — spur, bevel, helical, double helical, worm, and others — but in this chapter only, as previous defined, helical and double helical gears it will be considered. As gear teeth mesh, they roll and slide together. In the progression of contact as a pair of gear, the first contact is between a point near the root of the driving tooth and a point at the tip of the driven tooth. At this, the preceding teeth are still in mesh and carrying most of the load. As the contact progresses, the teeth roll and slide on each other from this position to the position where two pairs of teeth are engaged so that the transition of the load from the first to the next pair of teeth is made. Between these two positions the point of contact across through the particular position of the pitch point where there is pure rolling and/or no sliding. Having said this, it can be concluded that sliding varies from a maximum velocity in one direction at the start of mesh, through zero velocity at the pitch line, then again to a maximum velocity in the opposite direction at the ends of mesh; this, at the same time that rolling is continuous and added to the previous slid effect.

As stated above, in the EHD lubrication the elastic deformation of the bodies in contact occurs so that the lubricant film is formed. As such, the points of contact extend into a contact area. For abundant lubrication, as suggested in previous chapters, the oil lubricant must be injected to the full extent of the gear facewidth in the initial meshing zone, so that the full coverage of that contact area is guaranteed. In the case of industrial gears, the teeth are loaded sufficiently to ensure a progressive elastic deformation of the teeth along the meshing line. So, a successive convergent zone exists, pushing the injected oil from the beginning to the end of the engagement.

After understanding the formation of the fluid film in the convergent and after to be selected, in the previous Chapter 3, the most appropriate lubricant, it is necessary to uncover the method of lubrication. Due to its applicability to this gearbox, it has already been shown that the injected lubrication was selected; however, it is necessary to design and dimensioning its circulating system so that the lubricating oil should be delivered in the right quantities under certain operating and assembling conditions, to the elements that require it.

[1]or even EDL

Injection is a *reuse* method, where the lubricant leaving the elements is collected and recirculated to lubricate again. It has the advantage of conserving lubricant, minimizing waste, and helping control environmental pollution, [37]. Reuse methods generally include systems supplying with the following group of action.

Lube Oil Tanks Which stores the oil required for the operation of the mechanism. It is important to note that the size of the tank must be such that it can store 2 to 40 times the volume of oil that is needed for operation. In this way the shortage of lubricating fluid is avoided as the renewal of the oil is guaranteed, since it is not always the same part of the oil in operation; thus, it is also allowed to pass through all the constituent components of the circulating system and not only through the mechanical elements, such as gears and rolling bearings. Lastly, the reservoir should be large enough to ensure that oil velocity in it will be low and that the oil will have sufficient rest time to allow adequate separation of water and entrained solids.

Integrated Filtration Solutions Which use some method to remove suspended contamination from oil for either the protection of equipment (screening) or extending the life of the oil (cleaning). Much of the older equipment used large oil filtrations or did not use at all, but with the increase of the machinery demands was seen the importance of cleaning the oil to extend the life of its components.

Cooling Systems Which its capacity should be adequate to prevent oil temperature from rising above a safe maximum during the hottest conditions.

Oil Distribution Blocks Which provide a single supply line. With oil distribution blocks (or *distributor manifolds*) no relief valve is required, since the manifold's type valves automatically reset themselves and continue cycling as long as pressure is applied to them to the supply line.

Pumps and Valves Which create system flow. In other words, their function is to convert mechanical energy to fluid energy, with a combination of flow and pressure. The three most general design are: *gear*, *vane* and *piston* types. *Gear* pumps design is exclusively used in constant volume systems. Valves are generally used to control and direct the flow created by the pumps.

Condition Monitoring The two most common parameters to measure on circulating oil systems are oil temperature and oil level. Changes in normal operating conditions help to have a premature knowledge of a possible failure. The permanent monitoring of the oil level in the reservoir ensures the required permanence of fluid stationarity, so that the aforementioned particle separation effect occurs.

The object of study of this work is the dimensioning of the circulation system's part responsible for delivering the oil lubricant to the components inside the gearbox and not so much with the choice of these different action group; however, for a better understanding of the overall context of the next dimensioning, it was left a brief overview about them. For a more complete approach, I recommend [37].

Applied in a general sense, a tubular body primarily used to transport any commodity possessing flow characteristics such as those found in liquidis, gases, vapors, liquefied solids and fine powders, are in the American nomenclature named as *pipe* or simply *tubes*. Between these two terms there is no very rigid distinction. In general, the term *pipe* is used for ducts whose function is strictly to conduct fluids, on the other hand the term *tube* is used for ones that are supposed for other functions such as heat exchange —boiler coils, heat exchangers and others. Thus, given the applicability in the context of this work, from now on the term *pipe* shall be used. This terms suppose a rigid body, so for flexible pipes the most appropriate terms will be *hose*. It is called *piping* to a set of pipes and their multiple accessories. The need for piping, in the context of injected lubrication, arises from the fact that the point requiring control of friction and heat dissipation is far away at the storage site—reservoir. So, it can be said that in injected lubrication of mechanical components, the pipes work with free surface which deliver the fluid at a certain gearbox's point of interest.

5.2 Oil Circulating system conception

5.2.1 Pipe material selection

Although it is not a main group of components such as the transmission group (or gears) and the rolling bearings, the lubrication circulating system is extremely fundamental to the correct and long-lasting operation of the gearbox, and, for this reason, it also requires a somewhat careful selection of pipe's materials.

Using the available natural resources of the time, it is thought that the first pipes were made in bamboo. The first metal pipes were made of lead and bronze by the Greeks and Romans. Nowadays, a comprehensive list of the materials used to manufacture pipes should be considered. Some of the materials include concrete, glass, lead, brass, copper, plastic, aluminium, cast iron, carbon steel, and steel alloys. With such a broad range of materials available, the selection was restricted to the materials available by **Pinto & Cruz Tubagens e Sistemas S.A.** vendor. They offer a range of chooses which comprises: polyvinyl chloride (PVC), polypropylene (PP), polyethylene (PE), cross-linked polyethylene (PeX), galvanized iron, copper, non alloy steels,and stainless Steels.

Plastic pipes were first refused. Despite an increasingly improved performance of plastics, metal materials continue to offer a wider range of operating temperatures. In addition, even if low, the oil circulating system will be under pressure, so the wall thickness should be such as to support its effect. Since the plastic has mostly lower mechanical resistance, it would require a thicker wall, limiting the compactness of the circulating systems layout by requiring higher bending radius. Due to their low corrosion resistance, the possibility of using iron was also eliminated. Despite being oxidation corrosion resistant, copper has poor corrosion resistance by erosion effect, since the fluid will be excited of flow motion also copper has been eliminated. It should be noted that its specific weight is higher than all other options, this could lead to a heavier oil piping solution.

Finally, bearing in mind that wind turbine is expected to run for more than 20 years, a long-term choice has been made through the use of stainless steel pipes. With good machinability, high mechanical properties even with low wall thicknesses, high corrosion resistance in a wide range of temperatures and low coefficient of friction, this seemed to be the best cost-effective option. Considering the stainless steel pipes, it can be chosen seamed or seamless ones. Having in mind a pressure drop as low as possible in the overall pipesway, seamless tubes were selected.

5.2.2 Thread Ports

The threaded connections are one of the oldest port end assembly method used with pipe fittings. In small diameter pipes these connections are low-cost and easy to carry out, being the maximum typical nominal diameter of 2', [63].

Unlike tapered threads, parallel thread ports do not require sealing by the threads, being the seal obtained by other means, typically an elastomeric seal, Assembly/Installation of [64]. So, they only provide the resistance (holding power) for service pressure and sealing is made possible by sandwiching a O-ring in its cavity as shown in Fig. 5.1. When assembled properly, parallel thread ports provide the best leak-free port connection available, i.e. this type of thread is highly dependent on its assembly, and for this reason, it presents poor flexibility to mounting misalignment.

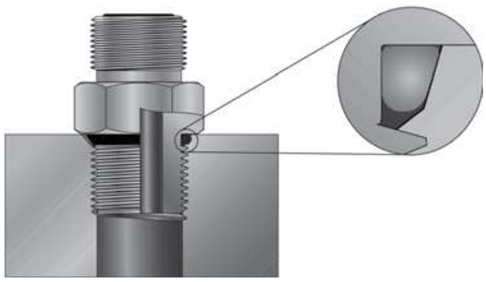


Figure 5.1: SAE / ISO / JIS B2351 Straight Thread Port O-Ring Upon Assembly. Source: Assembly/Installation section of [64].

Bearing in mind the aforementioned, tapered thread ports was chosen. In fact, there are various pipe fittings vendors; however, it makes little sense to order them from different sources and, as such, with a large range of selection PARKER® was chosen to be the sole supplier of pipe fittings for this application.

According to the pipe fittings made available by PARKER® , tapered thread ports include NPTF, BSPT and metric taper. The vast majority of Parker® pipe fittings division's standard pipe thread fittings are machined with the NPTF thread form; so, it was the preferred thread during the design process. In contrast to what happens with parallel threads, the tapered threads simultaneously serve to hold the fitting in place while under pressure and serve as the primary seal. Considering the mate between two NPTF threads, during the screwing, there is interference between the roots and crests of the male and female threads, ensuring the sealing, Fig. 5.2. This is a very specific case of tapered threads where full contact of the profiles gives a self-sealing property to the NPTF threads.

However, variations in condition of mating threads; inadequate assembly procedures; and, disturbances in operating conditions, like the high vibration level of the WT gearbox, make self-sealing highly improbable. So, for a higher safety from leakage a sealant shall be applied.

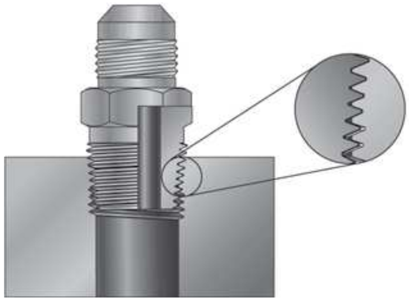


Figure 5.2: Tapered Thread Port. Source: Assembly/Installation section of [64].

5.2.3 Pipe End Assembly

In accordance to the section *Assembly/Installation* section of [64], the assembly of the pipe end consists of the following two steps:

1. Tube end preparation (cutting, deburring and cleaning)
2. Assembly and installation

Pipe end preparation is a very critical step to assure the integrity of a pipe assembly and an error may result in leakage. The preparation or the cutting tools of the different pipes will not be studied in this thesis; however, it is intended to leave a reference to the particularity of the steel pipes cut, namely the stainless steel ones. During the cut process, it is recurrent to create a large burr on the inner diameter, which may be difficult to correctly remove from this materials, creating flow restriction. Therefore, it is advisable to choose carefully the pipe cutting process.

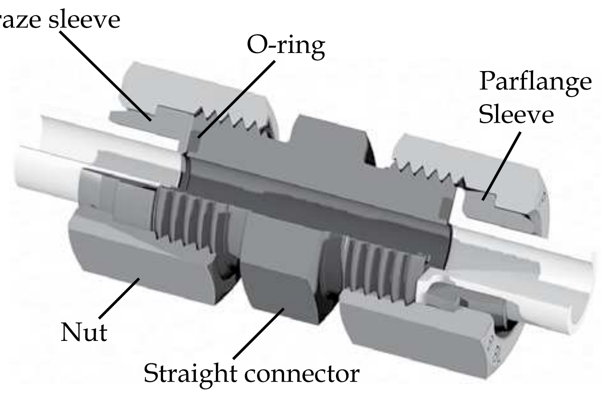


Figure 5.3: Sleeve attachment with flanged and brazed assemblies. Adapted from: Assembly/Installation section of [64].

This section is therefore intended to discuss one of the particular steps in the pipe end preparation process — the *Sleeve Attachment*. This can be accomplished by three methods: parflanging (or simply *flanging*), brazing and using a ferrule.

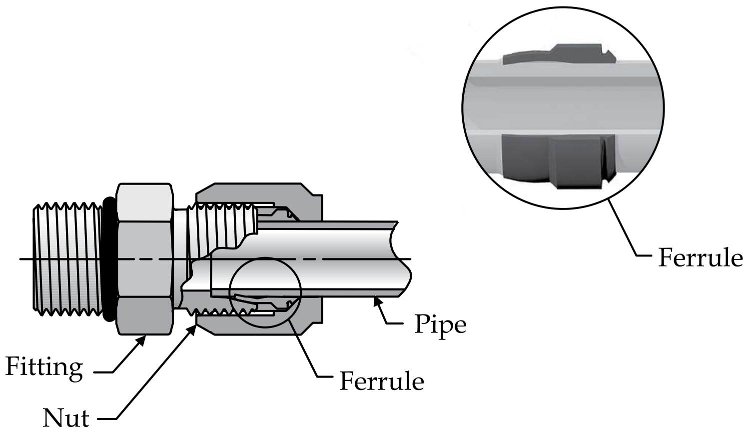


Figure 5.4: Sleeve attachment with ferrule assemblies. Adapted from: Assembly/Installation section of [64].

Parflange process is several times faster than the other two processes, especially than the braze one. It also does not require any special pre- or post-flange cleaning of the pipe and sleeve, like any brazing or welding process requires, or even requires any special presetting operation, like the ferrule process. Unlike brazing, the parflange process does not require any flux, braze alloy, post braze cleaner or rust inhibitor, being an environmentally safe lubricant the only additive associated with it. Safer than brazing without the need of an open flame or any form of heating, flanging process uses only a fraction of the energy needed for welding or brazing. The parflange process accommodates the exclusive use of pipe and sleeve, thus eliminating the need to an extra supporting sleeve typical in ferrule use. It also eliminates the potential for leaks, originated from a too large gap, result of a improper pipe cut (brazing) or a uneven bite (ferrule). For all these, and bearing in mind that parflange process produces a burnished sealing surface typically much smoother than the 125 micro-inch requirement of SAE J1453|[2], it was the sleeve attachment type selected.

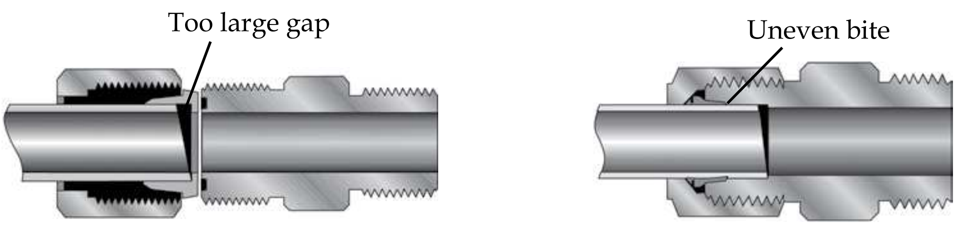


Figure 5.5: Sleeve attachment with ferrule assemblies. Adapted from: Assembly/Installation section of [64].

Some attention is required in the application of the flanging process. Two of the main precautions, namely what is related to the dimensioning of the pipes, are: the *extra cutt-off length* and the *minimum straight length to start of bend*. The first refers to the extra length, in relation to the effective flow pipe length, that the pipe must have in order to allow the flanging process. A wrong *extra cutt-off length* chosen can cause: **over-flanging**,

[2]Specification for O-Ring Face Seal Connectors, by SAE Interneational.

which will result in tube nut interference as well as thinning of the flange tube end; or, **under-flanging**, which reduces the contact area for sealing against the O-ring in the fitting. *Extra cutt-off length* and *minimum straight length to start of bend* recommended values should be consulted in Tables 5.1 and 5.2, respectively.

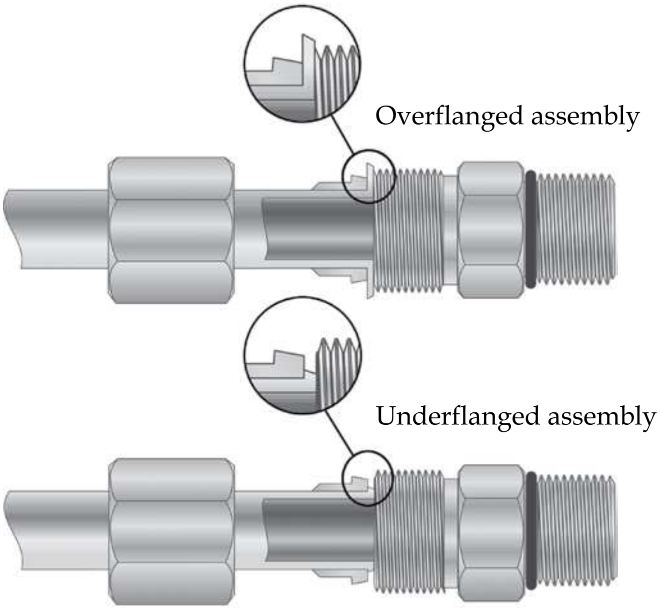


Figure 5.6: Over-flanging and Under-flanging failures. Source: Assembly/Installation section of [64].

Table 5.1: Extra tube cut-off length (use only as a guide), [65].

Pipe wall thickness	Metric Tube Outside Diameter (mm)															
mm	6	8	10	12	14	15	16	18	20	22	25	28	30	32	35	38
1.0	4.5	5.5	2.5	3.5	—	5.0										
1.5	5.5	5.0	4.0	4.5	4.5	4.5	3.0	6.0								
2.0			3.5	4.5	6.0	5.0	3.0	5.5	4.0	6.5	4.5	6.0	5.0	—	—	5.5
2.5				4.5	5.5	5.0	3.5	6.5	4.0	7.0	4.5	7.5	5.5	—	—	—
3.0							3.0	6.0	4.0	7.0	4.5	7.0	5.0	4.0	7.0	5.0
3.5									4.5	—	4.5	—	—	—	—	—
4.0									3.5	—	4.5	—	5.5	4.0	—	5.0
5.0											4.0	—	—	—	—	5.0

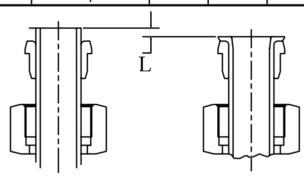
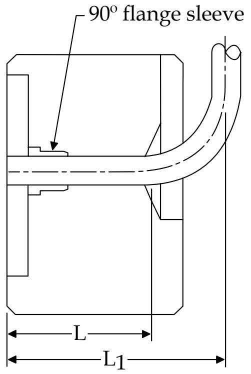


Table 5.2: Minimum straight length for 90°flanging, [65].



Pipe O.D. Metric	L mm	L1 mm
6	35	79
8	35	80
10	40	81
12	40	82
15	40	84
16	41	84
18	42	85
20	50	86
22	50	87
25	50	89
28	50	90
30	50	91
32	50	92
35	50	94
38	50	95

5.2.4 Pipe diameter selection

First and foremost to select piping diameters, it is prior to define the flow rates required by the components that need lubrication — gears, rolling bearings and seals. Consequently, this section is devoted to the analysis of the gearbox energy balance model, resulting in the statement of flow rates, pipe diameters, pipe wall thicknesses and minimum bending radius.

Among the dissipating energy flows, representative of the gearbox operation, are the energy dissipated in the gears of the WTG, which is created in the highly loaded contact between its teeth; the energy dissipated in the entire kinematics of the different rolling bearings; the dissipative energy in the contact between the seals lips and the rotating component; and etc... With this, energy evacuation mechanisms are needed, otherwise the temperature of the WTG would rise without restrictions, occurring, among other things, a severe pitting phenomenon in the mechanical components which could certainly lead to their total rupture. Among these evacuation mechanisms we mainly have the radiative, conductive and convective energy flow of the housing and the energy flow result of the interaction between the lubricant and the high temperature parts.

Bearing in mind that the efficiency of a gearbox is defined as the output and input power ratio, considering the above stated, this value will always be less than the unity. In the context of this work, the quantification of this value is linked to multiple parameters such as: type and weight of gears; number of transmission stages; rolling bearings selection; and, a few others. The dissipating energy flows mentioned above can be subdivided into two large groups: *load-influenced* power losses and *non load-influenced* power losses, [66].

Load-influenced power losses may be due to the friction during the meshing gear, which depends on the rated power, the average friction coefficient along the meshing line, and some geometric factors of the gear; or, due to the friction within the rolling bearings, which depends on rotational speed, friction coefficient, magnitude of the load and, again, geometric factors.

Non load-influenced power losses are in rolling bearings, seals and in the interaction with the oil. Regarding the interaction of the rotating components with the oil, there are frequent losses by churning, or, in other words, losses due to the friction result of the contact between the moving elements and the fluid in which they are partially dipped. Given the oil injection application, the oil level will have a residual height and so, it make any sense to refer this mechanism of energy dissipation. However, oil pumping losses or even internal losses in the circulating systems should be mentioned. Despite the existence of these losses, it will not be considered for the evacuation energy flows analysis; thus, it will be considered a parallel circuit relatively to the operation of the gearbox.

It should also be noted that in losses that are not influenced by the load, the losses in the seals are present.

For all that has been said and following the nomenclature proposed by [30], the overall power loss P_V of WTG is the sum of the power loss P_N, which depends on the load, and the loss of power P_L, dependent on the load,

$$P_V = P_N + P_L. \tag{5.1}$$

If we name P_Q the power which is evacuate through the housing, we set the thermal capacity by matching P_V e P_Q,

$$P_V = P_Q = A_C\ K\ \Delta T, \tag{5.2}$$

where, A_C is the area of the housing exposed to the air, K is the heat transfer coefficient and ΔT is the temperature difference between the oil and the ambient air. It must be noted that the heat transfer coefficient takes into account the three mechanisms of the housing thermal evacuation: *radiation*, *convection* and *conduction*.

The heat transfer coefficient of the gearbox housing, K, has a variable value, dependent on several factors such as its own geometry and the cooling effect from the outside. With no far information about the forced cooler usage inside the nacelle, analysing Tables 5.3 and 5.4, a heat transfer coefficient of 0.020 kW/(m^2 K) was chosen. This value ensure a balance between the two extreme situations presented in tables.

Table 5.3: Heat transfer coefficient without forced cooling, [30].

Condition	**Linear speed of the air** m/s	K kW/(m^2 K)
Small limited space	< 1.40	0.010–0.014
Great open space	< 1.40	0.016–0.020
Great open space	< 3.70	0.018–0.022
Exterior	> 3.70	0.020–0.025

Table 5.4: Heat transfer coefficient with forced cooling, [30].

Linear speed of the air m/s	K kW/(m^2 K)
2.5	0.015
5	0.024
10	0.042
15	0.058

Given a heat transfer coefficient of 0.020 kW/(m^2 K) and considering a 40 K between the oil and the ambient air, with a estimating shape housing like presented in Figure 5.7, which results in a total surface area of 18.057 m^2, it is obtained a power evacuation of about 14.446 kW, Table 5.5. This amount of power evacuation corresponds to approximately 45 % of the total sum of estimated losses in the contact between the teeth along meshing gear with the losses in all rolling bearing,which was calculated with the KISSsoft® calculation tool for the nominal load.

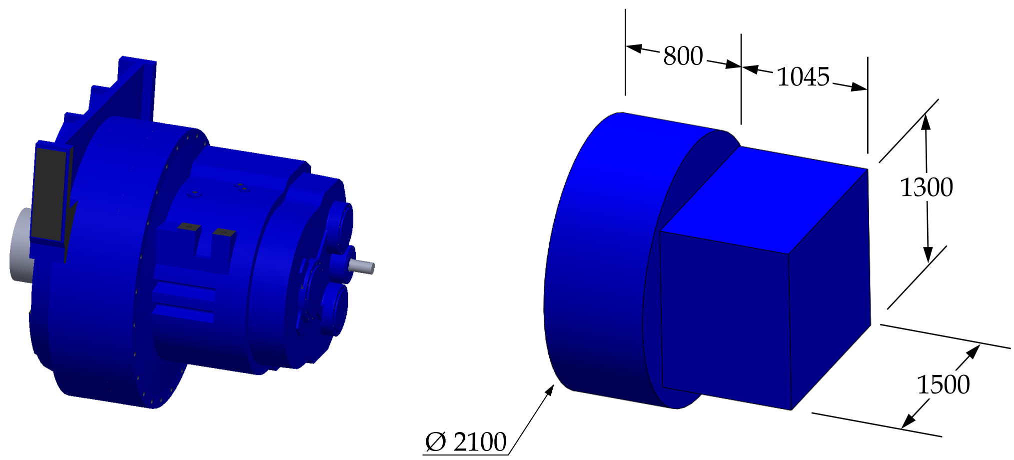


Figure 5.7: Estimated housing shape dimensions.

Table 5.5: Housing energy balance parameters and level of power evacuation.

Parameter	**Unit**	**Value**
total surface area, A$_c$	m^2	18.057
Heat transfer coefficient, K	**kW/(m^2 K)**	0.020
Temperature variation	**K**	40
Housing heat evacuation per second	**kW**	14.446

Looking at the percentage of power losses that would be possible to evacuate through gearbox housing is backed up the option of injected lubrication. The oil bath, as suggested in Section 3.1.11.4 of the II Part, would certainly not be enough to dissipate all the heat generated inside the WTG. In order to simplify the calculation, this dissipative effect of the housing will be neglected, being the piping dimensioning made based on the idea that all the heat generated must be evacuated through the interaction between the components and the oil lubricant. In this way and since we have an estimate of housing heat evacuation of about 45 %, not considering it we shall have a safety coefficient of **1.45**.

In addition, considering that the load spectrum proposed by the ISO 61400-4 standard includes a bin with a load factor of 1.67, an additional safety coefficient of **1.5** was considered. To summarize, relatively to the nominal load, the total oil required has a global safety coefficient of around **2.2**, which is extremely reasonable since the pressure drops along the circuit are being neglected.

Regarding the volume of oil, in the case of the gears, the amount of oil considered (in l/min) was required by the calculations of the KISSsoft® software. About the rolling bearings, there are no far information about flow rates required for each of the different types of rolling bearings and lubrication methods. Thus, following the recommended values of SKF® for a recirculating oil system (tab. 5.6) and considering a upper factor of 1.2 and 1.5 for bearings with asymmetric section with low and high power losses, respectively, the volume of oil obtained are shown in the Table 5.8. The **1.5** upper factor is due to the flow resistance that the rolling bearings may or may not offer. In fact the recommended values of oil volume suggested by SKF® do not distinguish the type of bearings; however, for rolling bearings with an asymmetrical cross section —such tapered roller bearings or spherical roller thrust bearings— larger flow rates are permissible than for rolling bearings with a symmetrical cross section, because their flow resistance is lower due to their pumping action. It was also considered an upper factor of **1.2** for the volume of oil of the rolling bearings that, despite the symmetrical section presented, a higher level of power losses is verified. This is due to the fact that higher power losses imply hotter runs and, consequently, more energy to be dissipated.

Table 5.6: Oil flow rate guidelines by SKF®, [67].

Bore diameter d^{BR}		**Bore diameter** Q	
mm		l/min	
over	incl.	low	high
—	50	0.3	1
50	120	0.8	3.6
120	400	1.8	6

In the Table 5.8, there are also shown the diameters of the pipes, which are responsible for the oil delivery to each gear pair as well as to each rolling bearing. The diameters $I.D.^{\mathrm{V}=2}$ were obtained through the equation,

$$I.D.^{\mathrm{V}=2} = \left(\frac{4}{\pi}\ \frac{60\text{x}10^3\ V_{oil}}{v_{oil}}\right)^{\frac{1}{2}}, \tag{5.3}$$

and assuming a linear speed of the oil lubricant v_{oil} of 2 m/s. In order to obtain a piping system according to the nominal pipe diameters available in the market, the $I.D.^{\mathrm{V}=2}$ diameters were converted to the final diameters $I.D.$. They are according to the size guide suggested by PARKER® in the table-5.7. In other words, assuming a wall thickness t close to the lowest values for each range, which is perfectly acceptable given the low pressure range of the circulating system, the diameters $O.D.$ was obtained selecting a outer diameter (among the PARKER suggested sizes) capable of covering the outer diameters $O.D.^{\mathrm{v}=2}$ calculated according to the following expression,

$$O.D.^{\mathrm{v}=2} = I.D.^{\mathrm{v}=2} + 2t. \tag{5.4}$$

Table 5.7: Recommended Min./Max. pipe wall thickness for metric system. Source: *Pipe Fittings and Port Adapters* section of [64].

	Steel, Alloy Steel, Stainless Steel, Copper, Monel	
O.D. Size in mm	**Wall Thickness in mm**	**Used With Fitting Size**
6	.5 - 2.25	-4
8	1.0 - 2.5	-6
10	1.0 - 3.0	-6
12	1.0 - 3.5	-8
14	1.0 - 4.0	-10
15	1.0 - 3.0	-10
16	1.0 - 3.0	-10
18	1.0 - 3.0	-12
20	1.5 - 4.0	-12
22	1.0 - 3.0	-16
25	2.0 - 5.0	-16
28	1.5 - 5.0	-20
30	2.0 - 5.0	-20
32	2.0 - 2.5	-20
35	2.0 - 6.0	-24
38	2.5 - 7.0	-24

5.2.5 Piping design

After all the diameters capable of meeting the oil volume requirements determined, the circulating oil system has been optimized with the goal of placing some lubrication path on the housing. This solution, in addition to a reduction in the number of piping accessories, it guarantees a reduction in threaded connections and ensure a better exploitation of the space, which promotes a solution as compact as possible.

Regarding meshing gears, oil delivery paths of the present gearbox are fully made by the use of pipes. Each pipe are equipped with a spray, that directs fresh oil at the convergent zone. The use of spray accessories provide a axial extended feed oil, which is need due to the high facewidth of the double helical gear. Although the single helical gear of the planetary stage, the rotary motion of the planet carrier does not allow a direct access to the consecutive points of engagement. Thus, the use of a spray injection makes it easier to extend the scope of the oil jet.

Table 5.8: Ultimate diameters of oil injection circulating system.

Port	**Designation**	**Flow rate, Q** l/min	$I.D.^{\mathbf{v=2}}$ mm	$I.D.$ mm	t mm	$O.D.$ mm	**Access. size**	**Min. bend. radius** mm
1	Main hose	265.314	51.779	58.2	2.9	64	32	XXX
2	SN/PL mesh and Carrier bearing	48.526	22.144	26.2	2.9	32	20	54
3	Planet bearings and 3rd superior mesh	68.928	26.392	26.2	2.9	32	20	54
4	2nd and 3rd stages' bearings	69	26.405	26.2	2.9	32	20	54
5	2nd and 3rd gear meshing's points	60.423	24.710	26.2	2.9	32	20	54
26	PL/RG mesh and Carrier bearing	18.437	13.650	16.7	1.65	20	12	30
13	(inf.) 2nd and 3rd gear meshing's points	39.926	20.582	21.7	1.65	25	16	37.5
27	All SN/PL meshes	38.026	20.087	24.7	1.65	28	20	42
31	All PL/RG meshes	7.937	9.177	16.7	1.65	20	16	30
6	Planet/Ring	2.646	6.118	9	1.5	12	8	18
7	Planet/Ring	2.646	6.118	9	1.5	12	8	18
8	Planet/Ring	2.646	6.118	9	1.5	12	8	18
9	All planet bearings	49.5	22.918	24.7	1.65	28	20	42
10	(sup.) 3rd gear meshing	19.428	14.357	16.7	1.65	20	12	30
11	(gen.) Carrier bearing	10.5	10.555	13	1.5	16	10	24
12	(sup.) 2nd gear meshing	20.498	14.747	16.7	1.65	20	12	30
14	(inf.) 2nd gear meshing	20.498	14.747	16.7	1.65	20	12	30
15	(inf.) 3rd gear meshing	19.428	14.357	16.7	1.65	20	12	30
17	(rotor) Sun shaft bearings	21	14.928	16.7	1.65	20	12	30
18	(rotor inf.) Int. bearing	6	7.979	9	1.5	12	8	18
19	(rotor sup.) Int. bearing	6	7.979	9	1.5	12	8	18
20	(rotor) High Speed bearing	4.5	6.910	9	1.5	12	8	18
21	(generator) Sun bearing	10.5	10.555	13	1.5	16	10	24
22	(generator sup.) Int. bearing	6	7.979	9	1.5	12	8	18
23	(generator inf.) Int. bearing	6	7.979	9	1.5	12	8	18
24	(generator) High Speed bearing	9	9.772	13	1.5	16	10	24
25	(rotor) Carrier bearing	10.5	10.555	13	1.5	16	10	24
27	PL/SN gear meshing	12.675	11.597	13	1.5	16	10	24
28	PL/SN gear meshing	12.675	11.597	13	1.5	16	10	24
29	PL/SN gear meshing	12.675	11.597	13	1.5	16	10	24

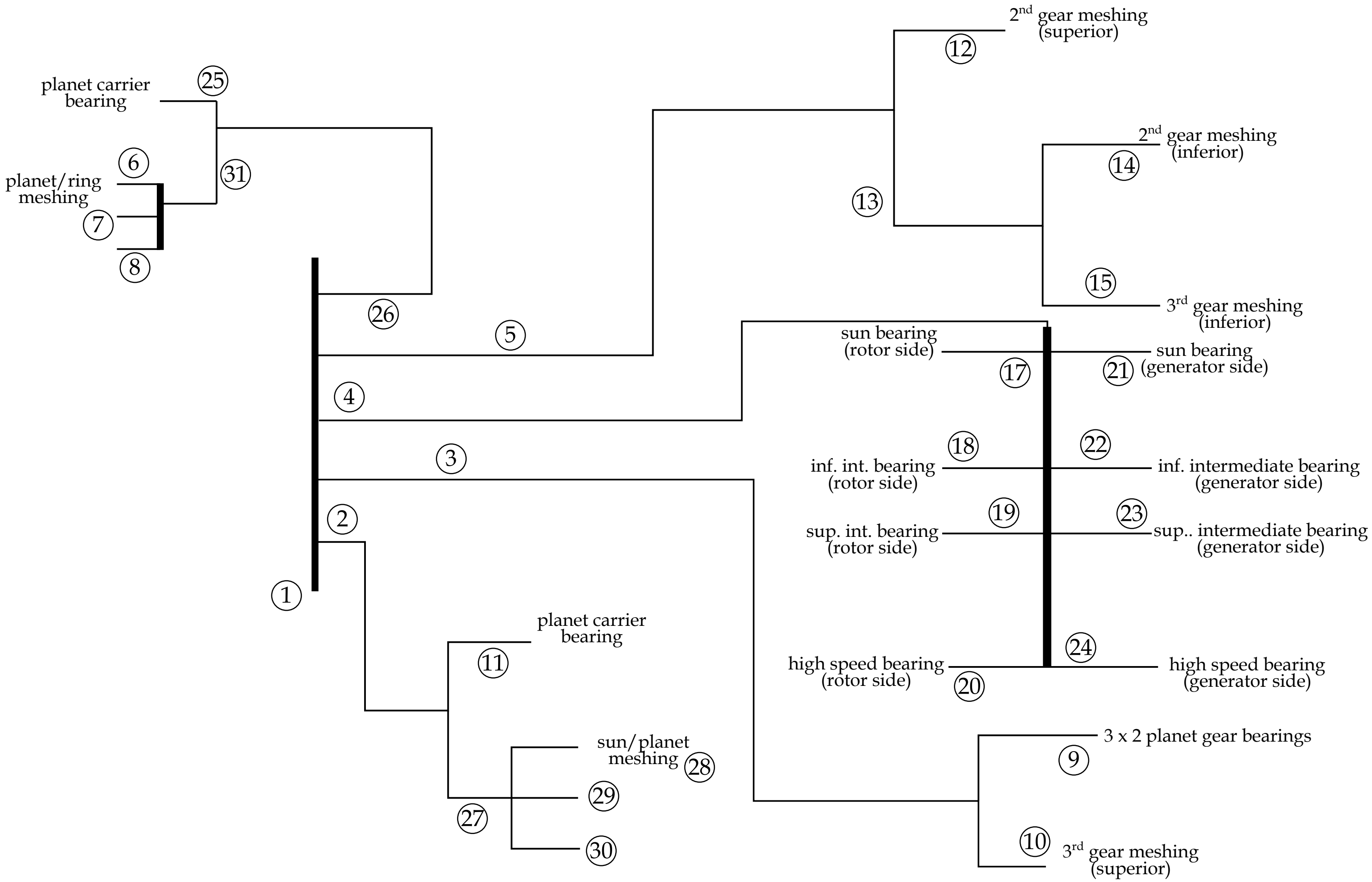


Figure 5.8: Circulating oil desgin.

Part IV

Chapter 6

Conclusion and future work

6.1 Conclusions

The main objective of this dissertation was the design of a gearbox for a 2 MW wind turbine capable of verifying the speed increase of its rotor at a speed compatible with the generation of energy by the generator.

It was started by a market study that allowed to situate us among the current practice, moving later on the kinematic analysis, calculation of resistance and life time of the gears and rolling bearings. The driver lines have always been to ensure the highest efficiency and the smallest size as much as possible.

After the remaining mechanical components were chosen, the gearbox thermal balance was analysed, allowing the dimensioning and design of a lubrication system. This includes piping, housing integrated lubrication paths and threaded fittings.

Finally, the assembly drawing was made, being susceptible to improvements after the public defense of this dissertation. It was expected to present the drawing of the main parts after all argued.

In the following topics, an overview of the various chapter and conclusions reached will be presented.

Chapter 3

This chapter was essentially devoted to the study of the chain of transmission. The criterion of the sum of profile shift coefficients always positive was followed for external gearing and preferred for internal gears. The preference for lower modules values was also preferred, minimizing the sliding effects and allowing a more efficient solutions.

In this chapter, a gearbox performing a three transmission stages layout was obtained. It has a first planetary stage and a double branch solution for the 2nd and 3rd transmission stages. The main advantage of the double branch was the possibility of deactivating one of the power paths for maintenance, keeping the remaining path, even below the full rated power, in operation.

Chapter 4

This chapter was mainly focused on the optimization of the solution obtained in the previous chapter, allowing small important adjustments.

With the application of the load spectrum there was essentially an impact on the life time of the rolling bearings, in the sense of increasing the longevity of their operation. With the application of modifications in the tooth profile a significant improvement in tooth resistance was observed, namely in the bending resistance in the root of the tooth. This was, in general, the limiting strength factor.

In this chapter, the application of mechanisms to relieve the stress concentrations phenomenon was also briefly discussed.

In addition, the particular assembly of the second part of this gearbox was treated. It should be noted that having the second and third transmission stages connected to the same input and output shafts, respectively, it can be said the gearbox mounting is in closed mesh, not allowing relative angular displacement between the power transmission elements of the same intermediate shaft. Here, it is proved that it is possible to determine the angular mismatch between the referred elements, being a well defined position provider an adequate assembly.

All the selection of rolling bearings, as well as, the application of spline joints was discussed in this chapter, aiming the pursuit of greater efficiency.

Chapter 5

In this part of the thesis was made a brief theorical introduction to what is lubrication, as well as, its applicability to the most important mechanical elements. It could be concluded that this is a EHD type of lubrication with a high level of heat generated. As a consequence an oil injection lubrication system was chosen instead of insufficient oil bath lubrication.

6.2 Future Work

Given the scale of the task proposed, a continuation of this work is needed to reach the proposed objective. The following list shows the main additions and changes needed to meet the objective:

- Although the current solution is satisfactory, different combinations of parameters must be developed and tested in the dimensioning of the transmission chain gears, which can improve their efficiency,
- A study with a broader spectrum could be carried out in order to avoid a somewhat oversized solution. Note that this was not possible given the high computational cost;
- Expand and verify the practical applicability of the calculated angular mismatch of the upper and lower intermediate gears. Not only the manufacturing implications of

this solution can be studied, as well as, the exploration of a mechanism that allows free angular displacement between the *second stage* pinion and the *third stage* gear, during assembly;

- The maximum deflection in gear zones was a bit higher than the typical hundredth of a module. This decision was essentially due to the importance of ensuring a somewhat floating assembly along the entire chain of transmission, since in all the transmission stages there is a split power, ideally, balanced. Note that the criteria of a *deflection-total length* ratio range of [0.001;0.003] was fully accomplished for all shafts. This leads to the suggestion of a more detailed analysis of these fluctuation intervals in which must be taken into account, for example, manufacturing deviations and deformation of the flanks teeth during the gear meshing;

- The amount of oil for lubricating the rolling bearings was made according to guidance values of SKF producer and it were considered upper factors for the rolling bearings with more critical behaviors. It is suggested to verify the sufficiency of these factors faces up the asymmetry characteristics of some types of rolling bearings;

- It is known that the storage of the oil in the bottom of a general gearbox can provide the increase of its temperature, as well as, the appearance of drag effects between the moving parts and the fluid; however, due to the typical low rotational speed of the planet carrier, which characterize the first transmission stage of the wind turbines, these effects were neglected and it was considered, even a small ones, a determined level of oil. It is therefore suggested that the appropriate residual oil level to be checked in the planetary gear;

- It would be interesting to carry out a study of the geometry and wall thickness of the housing, in sense of the best energy dissipation as well as its mass reduction;

- Finally, and with extreme relevance, it is suggest the analysis, parameterized study and design of a coupling system. It is known that dynamic and/or vibratory behavior in wind turbines is extremely critical contributing to high noise levels. Given the need for meticulous analysis for small improvements, this was not analyzed in this dissertation. However, it would be interesting as future work the conception of positioning mechanisms capable of supporting and control this dynamic behaviour.

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Appendix A

KISSsoft Reports

A.1 KISSsoft Reports - Gearing calculation

A.1.1 KISSsoft Report - Planetary stage

KISSsoft Release 03/2017 A

KISSsoft University license - Universidade do Porto

File

Name : Unnamed

Changed by: Joana Mêda de Sousa on: 24.09.2017 at: 05:11:03

Important hint: At least one warning has occurred during the calculation:

1-> Special load spectrum:
Elements with power = 0 or speed = 0 are not usual!
(Element no. 4)

CALCULATION OF A HELICAL PLANETARY GEAR STAGE

Drawing or article number:
Gear 1: 0.000.0
Gear 2: 0.000.0
Gear 3: 0.000.0

Load spectrum: Planet carrier

Own Input
Number of bins in the load spectrum: 11
Reference gear: Planet carrier

Bin No.	Frequency [%]	Power [kW]	Speed [1/min]	Torque [Nm]
1	0.00124	-1220.0004	15.0	-776676.4000
2	0.00103	-760.0003	15.0	-483831.2000
3	0.01066	-460.0002	15.0	-292845.2000
4	9.39362	0.0000	15.0	0.0000
5	27.24773	460.0002	15.0	292845.2000
6	19.30359	900.0003	15.0	572958.0000
7	13.09509	1360.0005	15.0	865803.2000
8	23.13569	1960.0007	15.0	1247775.2000
9	7.79235	2420.0009	15.0	1540620.4000
10	0.01894	2880.0010	15.0	1833465.6000
11	0.00006	3340.0012	15.0	2126310.8000

Bin No.	Coefficients KV	KHβ1	KHβ2	KHα1	KHα2	Kγ	YM1	YM2	YM3	Temperature Oiltemp
1	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000
2	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000
3	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000
4	1.0000	1.0000	1.0000	0.0000	0.0000	1.1000	1.0000	0.7000	1.0000	0.0000
5	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000
6	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000
7	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000
8	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000
9	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000

10	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000
11	1.0500	1.1500	1.1500	1.0000	1.0000	1.1000	1.0000	0.7000	1.0000	65.0000

Notice:
- Tooth flank with load spectrum: Check both cases and document the unfavorable case
- Tooth root with load spectrum: Check both cases and document the more realistic case (DIN3990-6, Method C)
  Is only applied on load spectrum bins, where the alternating bending factor (mean stress influence factor) YM=1.0.

S-N curve (Woehler line) in the endurance domain according: according to standard

Notice:
Calculation with methods ISO6336 and AGMA 2001 results in a reduction of resistance in the domain of fatigue resistance (from circa 10^7 to 10^10 cycles).
The lifetime calculation takes this into account (also with the S-N curve (Woehler Curve) of the Miner type).

Results

Safeties, calculated with load spectrum:

Root safety	2.295	1.566 / 1.867	2.234
Flank safety	1.380	1.462 / 3.082	2.489

Safeties against scuffing/micropitting/EHT/TFF are indicated for the most critical element of the load spectrum:

Scuffing safety (integral temperature)	3.251	5.022
Scuffing safety (flash temperature)	2.845	21.955
Safety against micropitting (B)	1.094	1.838

Safeties, calculated with nominal torque:

Safety against micropitting (B)	1.335	2.024

Analysis of critical elements in load spectrum: See section 10

ONLY AS INFORMATION: CALCULATION WITH REFERENCE POWER

Calculation method ISO 6336:2006 Method B

		------- Sun	----------- Planets	----------- Internal gear ---
Number of planets	[p]	(1)	3	(1)
Power (kW)	[P]		2000.001	
Speed (1/min)	[n]	77.9		0.0
Speed difference for planet bearing calculation (1/min)	[n2]		39.9	
Speed planet carrier (1/min)	[nSteg]		15.0	
Torque (Nm)	[T]	245216.6	0.0	1028023.4
Torque Pl.-Carrier (Nm)	[TSteg]		1273240.000	

Application factor	[KA]		1.00	
Distribution factor	[Kgam]		1.10	
Required service life (h)	[H]		175200.00	
Gear driving (+) / driven (-)		-	+/-	+

Working flank gear 1: Left flank
Sense of rotation gear 1 clockwise

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

		GEAR 1	GEAR 2	GEAR 3
Center distance (mm)	[a]		566.000	
Centre distance tolerance	ISO 286:2010 Measure js7			
Normal module (mm)	[mn]		16.0000	
Pressure angle at normal section (°)	[alfn]		20.0000	
Helix angle at reference circle (°)	[beta]		15.0000	
Number of teeth	[z]	26	41	-109
Facewidth (mm)	[b]	265.00	256.00	265.00
Hand of gear		left	right	right
Planetary axles can be placed in regular pitch.:		120°		
Accuracy grade	[Q-ISO1328:1995]	6	6	7
Inner diameter (mm)	[di]	0.00	0.00	
External diameter (mm)	[di]			1975.54
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00	
Outer diameter of gear rim (mm)	[dbi]			0.00

Material

Gear 1: 18CrNiMo7-6, Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC

Gear 2: 18CrNiMo7-6, Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC

Gear 3: 42 CrMo 4 (2), Through hardened steel, flame/ind. hardened
ISO 6336-5 Figure 11/12 (MQ) Flank & root hardened

		GEAR 1	GEAR 2	GEAR 3
Surface hardness		HRC 61	HRC 61	HRC 56
Material quality according to ISO 6336:2006 Normal (Life factors ZNT and YNT >=0.85)				
Fatigue strength. tooth root stress (N/mm²)	[σFlim]	500.00	500.00	370.00
Fatigue strength for Hertzian pressure (N/mm²)	[σHlim]	1500.00	1500.00	1220.00
Tensile strength (N/mm²)	[σB]	1200.00	1200.00	1100.00
Yield point (N/mm²)	[σS]	850.00	850.00	900.00
Young's modulus (N/mm²)	[E]	206000	206000	206000
Poisson's ratio	[v]	0.300	0.300	0.300
Roughness average value DS, flank (µm)	[RAH]	0.50	0.50	0.60
Roughness average value DS, root (µm)	[RAF]	3.00	3.00	3.00
Mean roughness height, Rz, flank (µm)	[RZH]	3.00	3.00	4.80
Mean roughness height, Rz, root (µm)	[RZF]	20.00	20.00	20.00

Gear reference profile 1 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000
	not topping	

Gear reference profile 2 :

Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A	
Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000
	not topping	

Gear reference profile 3 :

Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A	
Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000
	not topping	

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250	1.250
Tooth root radius Refer. profile	[rofP*]	0.380	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000	0.000

Type of profile modification: for uniform mesh

Tip relief (µm)	[Ca]	105.00	105.00	105.00

Lubrication type		Oil injection lubrication
Type of oil (Own input)		Oil: Castrol Optigear Synthetic X 320
Lubricant base		Synthetic oil based on Polyalphaolefin
Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	325.00
Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	34.90
Specific density at 15 °C (kg/dm³)	[roOil]	0.854
Oil temperature (°C)	[TS]	65.000

		------- GEAR 1 ------------ GEAR 2 ------------ GEAR 3 ---		
Overall transmission ratio	[itot]	0.193		
Gear ratio	[u]	1.577	-2.659	
Transverse module (mm)	[mt]	16.564		
Pressure angle at pitch circle (°)	[alft]	20.647		
Working transverse pressure angle (°)	[alfwt]	23.446	21.389	
	[alfwt.e/i]	23.455 / 23.438	21.380 / 21.398	
Working pressure angle at normal section (°)	[alfwn]	22.702	20.717	
Helix angle at operating pitch circle (°)	[betaw]	15.286	15.071	
Base helix angle (°)	[betab]	14.076		
Reference centre distance (mm)	[ad]	554.908	563.190	
Sum of profile shift coefficients	[Summexi]	0.7391	-0.1786	
Profile shift coefficient	[x]	0.3934	0.3457	-0.5244

Tooth thickness (Arc) (module) (module)	[sn*]	1.8572	1.8225	1.1891
Tip alteration (mm)	[k*mn]	-0.734	-0.734	0.000
Reference diameter (mm)	[d]	430.675	679.141	1805.522
Base diameter (mm)	[db]	403.013	635.521	1689.555
Tip diameter (mm)	[da]	473.795	720.736	1790.301
(mm)	[da.e/i]	473.795 / 473.785	720.736 / 720.726	1790.301 / 1790.311
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010	-0.000 / 0.010
Chamfer (1) / tip rounding (2: in transverse section, 3: in axial cross-section, 4: In normal section)		0	0	1
Tip chamfer (mm)	[hK]			0.000
Tooth tip chamfer angle (°)	[delhK]			45.000
Tip form diameter (mm)	[dFa]	473.795	720.736	1790.301
(mm)	[dFa.e/i]	473.795 / 473.785	720.736 / 720.726	1790.301 / 1790.311
Active tip diameter (mm)	[dNa.e/i]	473.795 / 473.785	720.736 / 720.726	1790.301 / 1790.311
Operating pitch diameter (mm)	[dw]	439.284	692.716 / 682.529	1814.529
(mm)	[dw.e]	439.311	692.759 / 682.487	1814.417
(mm)	[dw.i]	439.256	692.674 / 682.572	1814.642
Root diameter (mm)	[df]	403.263	650.204	1862.301
Generating Profile shift coefficient	[xE.e/i]	0.3822 / 0.3771	0.3307 / 0.3238	-0.5518 / -0.5630
Manufactured root diameter with xE (mm)	[df.e]	402.906	649.723	1863.180
(mm)	[df.i]	402.741	649.503	1863.537
Theoretical tip clearance (mm)	[c]	4.000	4.000/ 4.783	4.049
Tip clearance upper allowance (mm)	[c.e]	4.391	4.301/ 5.441	4.439
Tip clearance lower allowance (mm)	[c.i]	4.206	4.144/ 5.187	4.254
Active root diameter (mm)	[dNf]	417.875	666.636/660.319	1849.677
(mm)	[dNf.e]	417.927	666.695/660.379	1849.590
(mm)	[dNf.i]	417.828	666.583/660.267	1849.755
Root form diameter (mm)	[dFf]	414.478	660.546	1853.920
(mm)	[dFf.e/i]	414.243 / 414.135	660.175 / 660.007	1854.787 / 1855.138
Internal toothing: Calculation dFf with pinion type cutter (z0= 35, x0= 0.000)				
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	1.896 / 1.793	0.186 / 0.046	2.774 / 2.516
Addendum (mm)	[ha = mn * (haP*+x)]	21.560	20.797	7.610
(mm)	[ha.e/i]	21.560 / 21.555	20.797 / 20.792	7.610 / 7.605
Dedendum (mm)	[hf = mn * (hfP*-x)]	13.706	14.469	28.390
(mm)	[hf.e/i]	13.884 / 13.967	14.709 / 14.819	28.829 / 29.008
Roll angle at dFa (°)	[xsi_dFa.e/i]	35.417 / 35.415	30.649 / 30.647	20.079 / 20.080
Roll angle to dNf (°)	[xsi_dNf.e/i]	15.731 / 15.678	18.165 / 18.131	
	[xsi_dNf.e/i]		16.181 / 16.144	25.522 / 25.535
Roll angle at dFf (°)	[xsi_dFf.e/i]	13.619 / 13.553	16.114 / 16.057	25.952 / 25.980
Tooth height (mm)	[h]	35.266	35.266	36.000
Virtual gear no. of teeth	[zn]	28.609	45.115	-119.940
Normal tooth thickness at tip circle (mm)	[san]	10.630	11.905	13.548
(mm)	[san.e/i]	10.494 / 10.422	11.725 / 11.635	13.234 / 13.101
Normal space width at root circle (mm)	[efn]	12.893	11.965	8.740
(mm)	[efn.e/i]	0.000 / 0.000	12.025 / 12.053	8.680 / 8.655
Max. sliding velocity at tip (m/s)	[vga]	0.346	0.400/ 0.091	0.000
Specific sliding at the tip	[zetaa]	0.488	0.488/ 0.167	0.195
Specific sliding at the root	[zetaf]	-0.952	-0.952/ -0.242	-0.201
Sliding factor on tip	[Kga]	0.277	0.239/ 0.083	0.064
Sliding factor on root	[Kgf]	-0.239	-0.277/ -0.064	-0.083

Pitch on reference circle (mm)	[pt]		52.039			
Base pitch (mm)	[pbt]		48.696			
Transverse pitch on contact-path (mm)	[pet]		48.696			
Lead height (mm)	[pz]	5049.484		7962.647		21168.989
Axial pitch (mm)	[px]	194.211		194.211		194.211
Length of path of contact (mm)	[ga]		69.336		80.350	
(mm)	[ga.e/i]		69.424 / 69.228		80.446 / 80.229	
Length T1-A (mm)	[T1A]	124.562		100.644/169.980		376.400
Length T1-B (mm)	[T1B]	103.922		121.284/138.326		344.746
Length T1-C (mm)	[T1C]	87.393		137.813/124.459		330.878
Length T1-D (mm)	[T1D]	75.865		149.341/121.284		327.703
Length T1-E (mm)	[T1E]	55.226		169.980/89.630		296.049
Diameter of single contact point B (mm)	[d-B]	453.452		680.240/ 693.126		1824.827
(mm)	[d-B.e]	430.630		702.134/ 680.240		1812.293
(mm)	[d-B.i]	430.623		702.292/ 680.232		1812.147
Diameter of single contact point D (mm)	[d-D]	430.630		702.209/ 680.240		1812.224
(mm)	[d-D.e]	453.371		680.240/ 693.049		1824.827
(mm)	[d-D.i]	453.542		680.232/ 693.215		1824.839

Transverse contact ratio	[eps_a]	1.424	1.650
Transverse contact ratio with allowances	[eps_a.e/i]	1.426 / 1.422	1.652 / 1.648
Overlap ratio	[eps_b]	1.318	1.318
Total contact ratio	[eps_g]	2.742	2.968
Total contact ratio with allowances	[eps_g.e/i]	2.744 / 2.740	2.970 / 2.966

2. FACTORS OF GENERAL INFLUENCE

------- GEAR 1 ------------ GEAR 2 ------------ GEAR 3 ---

Nominal circum. force at pitch circle (N)	[Ft]		379585.000		379585.000	
Axial force (N)	[Fa]	101709.5		101709.5		101709.5
Axial force (total) (N)	[Fatot=Fa* 3]					305128.5
305128.5						
Radial force (N)	[Fr]		143031.315		143031.315	
Normal force (N)	[Fnorm]	418195.6		418195.6		418195.6
Nominal circumferential force per mm (N/mm)	[w]		1482.75		1482.75	
Only as information: Forces at operating pitch circle:						
Nominal circumferential force (N)	[Ftw]		372146.229		377700.650	
Axial force (N)	[Fa]	101709.5		101709.5/101709.5		101709.5
Axial force (total) (N)	[Fatot=Fa* 3]					305128.5
305128.5						
Radial force (N)	[Fr]		161399.856		147935.621	
Circumferential speed reference circle (m/s)	[v]			1.42 (Planet)		

Running-in value (µm)	[yp]	1.125	1.823
Running-in value (µm)	[yf]	1.500	2.550
Gear body coefficient	[CR]	1.000	1.000
Correction coefficient	[CM]	0.800	0.800
Basic rack factor	[CBS]	0.975	0.975
Material coefficient	[E/Est]	1.000	1.000
Singular tooth stiffness (N/mm/µm)	[c']	14.270	15.149
Meshing stiffness (N/mm/µm)	[cgalf]	18.806	22.535
Meshing stiffness (N/mm/µm)	[cgbet]	15.985	19.155
Reduced mass (kg/mm)	[mRed]	0.2049	1.6808
Resonance speed (min-1)	[nE1]	3518	853
Resonance ratio (-)	[N]	0.018	0.047

User specified factor KV:				
Dynamic factor	[KV]		1.05	1.05
User specified factor KHb:				
Face load factor - flank	[KHb]		1.15	1.15
- Tooth root	[KFb]		1.13	1.13
- Scuffing	[KBb]		1.15	1.15
User specified factor KHa:				
Transverse load factor - flank	[KHa]		1.00	1.00
- Tooth root	[KFa]		1.00	1.00
- Scuffing	[KBa]		1.00	1.00
Helical load factor scuffing	[Kbg]		1.26	1.28
Number of load cycles (in mio.)	[NL]	1983.1	419.2	473.0

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B
Internal toothing: Calculation of roF and sFn according to ISO 6336-3:2007-04-01
Internal toothing: Calculation of YF, YS with pinion type cutter (z0=35, x0= 0.000, rofP*= 0.380)

		------- GEAR 1	------------ GEAR 2	------------ GEAR 3 ---
Calculated with profile shift	[x]	0.3934	0.3457	-0.5244
Tooth form factor	[YF]	1.25	1.30/ 0.96	0.90
Stress correction factor	[YS]	2.20	2.19/ 2.48	2.42
Bending moment arm (mm)	[hF]	16.73	17.78/ 13.00	18.50
Working angle (°)	[alfFen]	23.18	22.35/ 20.43	21.37
Tooth thickness at root (mm)	[sFn]	35.40	36.01/ 36.01	44.27
Tooth root radius (mm)	[roF]	7.05	7.00/ 7.00	7.95

(hF* = 1.046/ 1.111/ 0.812/ 1.156 sFn* = 2.213/ 2.251/ 2.251/ 2.767)
(roF* = 0.441/ 0.437/ 0.437/ 0.497 dsFn = 408.908/656.123/656.123/ -1860.406 alfsFn = 30.0/ 30.0/ 30.0/ 60.0)

Helix angle factor	[Ybet]	0.87	0.87	
Deep tooth factor	[YDT]	1.00	1.00	
Gear rim factor	[YB]	1.00	1.00	1.00
Effective facewidth (mm)	[beff]	265.00	256.00/ 256.00	265.00
Nominal stress at tooth root (N/mm²)	[sigF0]	216.25	229.92/ 192.89	170.37
Tooth root stress (N/mm²)	[sigF]	281.80	299.62/ 251.37	222.01
Permissible bending stress at root of Test-gear				
Notch sensitivity factor	[YdrelT]	1.000	1.001/ 1.001	1.003
Surface factor	[YRrelT]	0.957	0.957	0.957
size factor (Tooth root)	[YX]	0.890	0.890	0.890
Finite life factor	[YNT]	0.878	0.906	0.904
Alternating bending factor (mean stress influence coefficient)	[YM]	1.000	0.700	1.000
Stress correction factor	[Yst]		2.00	
Yst*sigFlim (N/mm²)	[sigFE]	1000.00	1000.00	740.00
Permissible tooth root stress (N/mm²)	[sigFP=sigFG/SFmin]	479.29	346.31/ 346.31	365.92
Limit strength tooth root (N/mm²)	[sigFG]	747.68	540.25/ 540.25	570.84
Required safety	[SFmin]	1.56	1.56	1.56

4. SAFETY AGAINST PITTING (TOOTH FLANK)

		GEAR 1	GEAR 2	GEAR 3
Zone factor	[ZH]		2.26	2.38
Elasticity factor (√N/mm²)	[ZE]		189.81	189.81
Contact ratio factor	[Zeps]		0.838	0.778
Helix angle factor	[Zbet]		1.017	1.017
Effective facewidth (mm)	[beff]		256.00	256.00
Nominal contact stress (N/mm²)	[sigH0]		867.67	417.31
Contact stress at operating pitch circle (N/mm²)	[sigHw]		999.99	480.95
Single tooth contact factor	[ZB,ZD]	1.00	1.00/ 1.00	1.00
Contact stress (N/mm²)	[sigHB, sigHD]	999.99	999.99/ 480.95	480.95
Lubrication coefficient at NL	[ZL]	1.048	1.048/ 1.048	1.048
Speed coefficient at NL	[ZV]	0.959	0.959/ 0.959	0.959
Roughness coefficient at NL	[ZR]	1.046	1.046/ 1.061	1.061
Material pairing coefficient at NL	[ZW]	1.000	1.000/ 1.000	1.000
Finite life factor	[ZNT]	0.893	0.937	0.933
Limited pitting is permitted:	No			
Size factor (flank)	[ZX]	1.000	1.000	1.000
Permissible contact stress (N/mm²)	[sigHP=sigHG/SHmin]	1127.02	1182.05/ 1198.85	971.46
Pitting stress limit (N/mm²)	[sigHG]	1408.78	1477.56/ 1498.56	1214.32
Required safety	[SHmin]	1.25	1.25	1.25

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Pairing Gear 1- 2:

Calculation of permissible specific film thickness

Lubricant load according to FVA Info sheet 54/7 10 (Oil: Castrol Optigear Synthetic X 320)

Reference data FZG-C Test:

Torque (Nm)	[T1Ref]	265.1
Line load at contact point A (N/mm)	[FbbRef,A]	236.3
Oil temperature (°C)	[theOilRef]	60.0
Tooth mass temperature (°C)	[theMRef]	90.3
Contact temperature (°C)	[theBRef,A]	181.8
Lubrication gap thickness (µm)	[hRef,A]	0.130
Specific film thickness in test (µm)	[lamGFT]	0.260
Material coefficient	[WW]	1.00
Permissible specific film thickness (µm)	[lamGFP]	0.363

Interim results in accordance with ISO/TR 15144:2014

Coefficient of friction	[mym]	0.041
Lubricant factor	[XL]	0.800
Roughness factor	[XR]	0.679
Tooth mass temperature (°C)	[theM]	67.1
Tip relief factor	[XCa (A)]	1.597
Loss factor	[HV]	0.121
Equivalent Young's modulus (N/mm²)	[Er]	226374
Pressure-viscosity coefficient (m²/N)	[alf38]	0.01380
Dynamic viscosity (Ns/m²)	[etatM]	81.3
Roughness average value (µm)	[Ra]	0.5

Calculation of speeds, load distribution and flank curvature according to method B following ISO/TR 15144-1:2014

Ca taken as optimal in the calculation (0=no, 1=yes) 1 1

Pairing Gear 2- 3:

Calculation of permissible specific film thickness

Material coefficient	[WW]	0.82

Permissible specific film thickness (µm) [lamGFP] 0.300

Interim results in accordance with ISO/TR 15144:2014
Coefficient of friction [mym] 0.024
Lubricant factor [XL] 0.800
Roughness factor [XR] 0.500
Tooth mass temperature (°C) [theM] 66.0
Tip relief factor [XCa (A)] 1.000
Loss factor [HV] 0.036
Equivalent Young's modulus (N/mm²) [Er] 226374
Pressure-viscosity coefficient (m²/N) [alf38] 0.01380
Dynamic viscosity (Ns/m²) [etatM] 84.9
Roughness average value (µm) [Ra] 0.6
Calculation of speeds, load distribution and flank curvature according to method B following ISO/TR 15144-1:2014
Ca taken as optimal in the calculation (0=no, 1=yes) 1 1

5. SCUFFING LOAD CAPACITY

Calculation method according to ISO TR 13989:2000

Lubrication coefficient (for lubrication type) [XS] 1.200
Scuffing test and load stage [FZGtest] FZG - Test A / 8.3 / 90 (ISO 14635 - 1) 13
Multiple meshing factor [Xmp] 2.0 2.0
Relative structure coefficient (Scuffing) [XWrelT] 1.000 1.000
Thermal contact factor (N/mm/s^.5/K) [BM] 13.780 13.780 13.780
Relevant tip relief (µm) [Ca] 105.00 105.00 105.00
Optimal tip relief (µm) [Ceff] 86.73 72.38
Ca taken as optimal in the calculation (0=no, 1=yes) 1 1/ 1 1
Effective facewidth (mm) [beff] 256.000 256.000
Applicable circumferential force/facewidth (N/mm) [wBt] 1969.468 1969.468
(1) Kbg = 1.259, wBt*Kbg = 2479.319
(2) Kbg = 1.281, wBt*Kbg = 2521.927
Angle factor [Xalfbet] 1.021 0.991

Flash temperature-criteria
Lubricant factor [XL] 0.639 0.639
Tooth mass temperature (°C) [theMi] 85.32 67.65
(theMi = theoil + XS*0.47*Xmp*theflm)
Average flash temperature (°C) [theflm] 18.01 2.35
Scuffing temperature (°C) [theS] 323.17 323.17
Coordinate gamma (point of highest temp.) [Gamma] 0.305 0.242
(1) [Gamma.A]=0.425 [Gamma.E]=-0.368
(2) [Gamma.A]=0.366 [Gamma.E]=-0.280
Highest contact temp. (°C) [theB] 120.75 72.22
Flash factor (°K*N^-.75*s^.5*m^-.5*mm) [XM] 50.058 50.058
Approach factor [XJ] 1.000 1.000
Load sharing factor [XGam] 0.825 0.735
Dynamic viscosity (mPa*s) [etaM] 88.12 88.12 (65.0 °C)
Coefficient of friction [mym] 0.057 0.046

Integral temperature-criteria
Lubricant factor [XL] 0.800
Tooth mass temperature (°C) [theMC] 78.07 66.02
(theMC = theoil + XS*0.70*theflaint)
Mean flash temperature (°C) [theflaint] 7.78 0.61
Integral scuffing temperature (°C) [theSint] 342.30 342.30

Flash factor (°K*N^-.75*s^.5*m^-.5*mm)	[XM]	50.058	50.058
Running-in factor (well run in)	[XE]	1.000	1.000
Contact ratio factor	[Xeps]	0.284	0.235
Dynamic viscosity (mPa*s)	[etaOil]	88.12	88.12
Mean coefficient of friction	[mym]	0.041	0.024
Geometry factor	[XBE]	0.245	0.047
Meshing factor	[XQ]	1.000	1.000
Tip relief factor	[XCa]	1.597	1.845
Integral tooth flank temperature (°C)	[theint]	89.75	66.94

6. MEASUREMENTS FOR TOOTH THICKNESS

		------- GEAR 1 ------------	GEAR 2 ------------	GEAR 3 ---
Tooth thickness deviation	DIN 3967 cd25	DIN 3967 cd25		DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.130/ -0.190	-0.175/ -0.255	-0.320/ -0.450
Number of teeth spanned	[k]	4.000	6.000	-0.000
(Internal toothing: k = (Measurement gap number)				
Base tangent length (no backlash) (mm)	[Wk]	176.057	273.714	-0.000
Actual base tangent length ('span') (mm)	[Wk.e/i]	175.935/175.878	273.549/273.474	-0
.000/	-0.000			
Diameter of contact point (mm)	[dMWk.m]	437.644	688.673	-0.000
Theoretical diameter of ball/pin (mm)	[DM]	29.430	28.203	26.867
Effective diameter of ball/pin (mm)	[DMeff]	30.000	30.000	28.000
Radial single-ball measurement backlash free (mm)	[MrK]	243.554	367.812	890.728
Radial single-ball measurement (mm)	[MrK.e/i]	243.420/243.358	367.617/367.527	891
.160/	891.336			
Diameter of contact point (mm)	[dMMr.m]	443.825	693.032	1820.432
Diametral measurement over two balls without clearance (mm)	[MdK]	487.107	735.106	1781
.268				
Diametral two ball measure (mm)	[MdK.e/i]	486.840/486.717	734.716/734.537	1782
.133/	1782.483			
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	486.840/486.717	735.233/735.054	-0
.000/	-0.000			
Measurement over 2 pins (free) according to AGMA 2002 (mm)				
	[dk2f.e/i]	0.000/ 0.000	734.672/734.494	0.000/ 0
.000				
Measurement over 2 pins (transverse) according to AGMA 2002 (mm)				
	[dk2t.e/i]	0.000/ 0.000	735.742/735.563	0.000/ 0
.000				
Measurement over 3 pins (axial) according to AGMA 2002 (mm)				
	[dk3A.e/i]	486.840/486.717	735.233/735.054	-0
.000/	-0.000			
Effective dimensions over 3 pins (mm)	[Md3R.e/i]	0.000/ 0.000	0.000/ 0.000	-0.000/ -0.000
Note: Internal gears with helical teeth cannot be measured with rollers.				
Tooth thickness (chordal) in pitch diameter (mm)	[sc]	29.694	29.151	19.025
(mm)	[sc.e/i]	29.564/ 29.504	28.976/ 28.896	18.705/ 18
.575				
Reference chordal height from da.m (mm)	[ha]	22.036	21.087	7.561
Tooth thickness (Arc) (mm)	[sn]	29.715	29.159	19.026
(mm)	[sn.e/i]	29.585/ 29.525	28.984/ 28.904	18.706/ 18
.576				
Backlash free center distance (mm)	[aControl.e/i]	565.628/ 565.457	566.655/ 566.931	
Backlash free center distance, allowances (mm)	[jta]	-0.372/ -0.543	0.655/ 0.931	

dNf.i with aControl (mm)	[dNf0.i]	417.159	658.958	1851.750
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	1.458	-0.609	1.518
Tip clearance (mm)	[c0.i(aControl)]	3.698	3.636	3.358
Centre distance allowances (mm)	[Aa.e/i]	0.035/ -0.035	-0.035/ 0.035	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.030/ -0.030	0.027/ -0.027	
Radial clearance (mm)	[jrw]	0.578/ 0.337	0.966/ 0.620	
Circumferential backlash (transverse section) (mm)	[jtw]	0.500/ 0.292	0.761/ 0.488	
Normal backlash (mm)	[jnw]	0.454/ 0.265	0.691/ 0.443	

Torsional angle at entry with fixed output:

Entire torsional angle (°)	[j.tSys]	0.0610/ 0.0424

7. GEAR ACCURACY

		------- GEAR 1	------------ GEAR 2	------------ GEAR 3 ---
According to ISO 1328-1:1995, ISO 1328-2:1997				
Accuracy grade	[Q]	6	6	7
Single pitch deviation (μm)	[fptT]	14.00	16.00	26.00
Base circle pitch deviation (μm)	[fpbT]	13.10	15.00	24.30
Sector pitch deviation over k/8 pitches (μm)	[Fpk/8T]	26.00	32.00	66.00
Profile form deviation (μm)	[ffaT]	18.00	20.00	34.00
Profile slope deviation (μm)	[fHaT]	15.00	16.00	28.00
Total profile deviation (μm)	[FaT]	23.00	26.00	44.00
Helix form deviation (μm)	[ffbT]	18.00	18.00	28.00
Helix slope deviation (μm)	[fHbT]	18.00	18.00	28.00
Total helix deviation (μm)	[FbT]	25.00	26.00	40.00
Total cumulative pitch deviation (μm)	[FpT]	50.00	64.00	133.00
Runout (μm)	[FrT]	40.00	51.00	107.00
Single flank composite, total (μm)	[FisT]	72.00	86.00	172.00
Single flank composite, tooth-to-tooth (μm)	[fisT]	22.00	22.00	39.00
Radial composite, total (μm)	[FidT]	110.00	121.00	201.00
Radial composite, tooth-to-tooth (μm)	[fidT]	68.00	68.00	97.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality) 6)

Maximum value for deviation error of axis (μm)	[fSigbet]	16.90	16.90
Maximum value for inclination error of axes (μm)	[fSigdel]	33.80	33.80

8. ADDITIONAL DATA

Mass (kg)	[m]	313.397	739.722	924 .625
Total mass (kg)	[m]	3457.187		

Moment of inertia (system with reference to the drive):
calculation without consideration of the exact tooth shape

single gears ((da+df)/2...di) (kg*m²)	[TraeghMom]	7.53360	43.44652	836.56902
System ((da+df)/2...di) (kg*m²)	[TraeghMom]	2371.79694		

Torsional stiffness at entry with driven force fixed:

Torsional stiffness (MNm/rad)	[cr]	6590.724	
Torsion when subjected to nominal torque (°)	[delcr]	0.009	
Mean coeff. of friction (acc. Niemann)	[mum]	0.039	0.028
Wear sliding coef. by Niemann	[zetw]	0.694	0.302

Meshpower (kW)		1614.815	1614.815
Gear power loss (kW)		2.563	0.548
Total power loss (kW)		9.334	
Total efficiency		0.995	
Sound pressure level (according to Masuda)	[dB(A)]	106.2	106.8

Classification according to F.E.M. (Edition 1.001, 1998)		
Spectrum factor	[km]	0.090
Spectrum class	[L]	1
Application class (predefined service life)	[T]	9
Machine class (predefined service life)	[M]	8
Application class (achievable service life)	[T]	9
Machine class (achievable service life)	[M]	8

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Profile and tooth trace modifications for gear 1

Symmetric (both flanks)

- Tip relief, linear Caa = 105.000µm LCa = 0.787*mn dCa = 461.053mm

Profile and tooth trace modifications for gear 2

Symmetric (both flanks)

- Tip relief, linear Caa = 105.000µm LCa = 1.090*mn dCa = 704.954mm

Profile and tooth trace modifications for gear 3

Symmetric (both flanks)

- Tip relief, linear Caa = 105.000µm LCa = 1.329*mn dCa =-1804.816mm

Tip relief verification							
Diameter (mm)	[dcheck]	473.465		720.406		-1790.631	
Tip relief left/right (µm)	[Ca L/R]	102.4 /	102.4	102.9 /	102.9	102.6 /	102.6

Data for the tooth form calculation :
Data not available.

10. SERVICE LIFE, DAMAGE

Calculation with load spectrum		
Required safety for tooth root	[SFmin]	1.56
Required safety for tooth flank	[SHmin]	1.25

Service life (calculated with required safeties):		
System service life (h)	[Hatt]	211782

Tooth root service life (h)	[HFatt]	1e+006	2.118e+005	1e+006
Tooth flank service life (h)	[HHatt]	1e+006	1e+006	1e+006

Note: The entry 1e+006 h means that the Service life > 1,000,000 h.

Damage calculated on the basis of the required service life (175200.0 h)

No.	F1%	F2%	F3%	H1%	H2%	H3%
1	0.00	0.00	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00	0.00	0.00

5	0.00	0.00	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00	0.00	0.00
9	0.00	76.57	0.00	3.53	0.75	0.00
10	0.00	6.11	0.00	0.15	0.03	0.00
11	0.00	0.05	0.00	0.00	0.00	0.00
Σ	0.00	82.73	0.00	3.68	0.78	0.00

Damage calculated on basis of system service life [Hatt] (211782.0 h)

No.	F1%	F2%	F3%	H1%	H2%	H3%
1	0.00	0.00	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00	0.00	0.00
9	0.00	92.55	0.00	4.27	0.90	0.00
10	0.00	7.39	0.00	0.18	0.04	0.00
11	0.00	0.06	0.00	0.00	0.00	0.00
Σ	0.00	100.00	0.00	4.45	0.94	0.00

Damage calculated on basis of individual service life HFatt & HHatt

	HFatt1	HFatt2	HFatt3	HHatt1	HHatt2	HHatt3	
(h)	1e+006	2.118e+005	1e+006		1e+006	1e+006	1e+006
No.	F1%	F2%	F3%	H1%	H2%	H3%	
1	0.00	0.00	0.00	0.00	0.00	0.00	
2	0.00	0.00	0.00	0.00	0.00	0.00	
3	0.00	0.00	0.00	0.00	0.00	0.00	
4	0.00	0.00	0.00	0.00	0.00	0.00	
5	0.00	0.00	0.00	0.00	0.00	0.00	
6	0.00	0.00	0.00	0.00	0.00	0.00	
7	0.00	0.00	0.00	0.00	0.00	0.00	
8	0.00	0.00	0.00	0.00	0.00	0.00	
9	0.00	92.55	0.00	95.94	95.94	0.00	
10	0.00	7.39	0.00	3.98	3.98	0.00	
11	100.00	0.06	100.00	0.08	0.08	0.00	
Σ	100.00	100.00	100.00	100.00	100.00	0.00	

Most critical duty cycle elements for Scoring (SB, Sint), Tooth Flank Fracture (SFF), hardened layer (SEHT) and Micropitting (Slam)

Calculation of the factors required to define reliability R(t) according to B. Bertsche with Weibull distribution; t in (h):
R(t) = 100 * [Exp(-((t*fac - t0)/(T - t0))^b)]^p %

Gear		p	fac	b	t0	T	R(H)%
1	Tooth root	1	11319	1.7	8.662e+015	1.331e+016	100.00
1	Tooth flank	1	11319	1.3	4.858e+010	2.315e+011	100.00
2	Tooth root	3	2393	1.7	4.892e+008	7.518e+008	100.00
2	Tooth flank	3	2393	1.3	4.858e+010	2.315e+011	100.00
3	Tooth root	1	2700	1.7	4.31e+014	6.624e+014	100.00
3	Tooth flank	1	2700	1.3	9.014e+029	4.295e+030	100.00

Reliability of the configuration for required service life (%) 100.00 (Bertsche)

REMARKS:

- Specifications with [.e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances
  Specifications with [.m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances
  The calculation is done for the operating pitch circle.
- Calculation of Zbet according Corrigendum 1 ISO 6336-2:2008 with Zbet = 1/(COS(beta)^0.5)
- Details of calculation method:
  cg according to method B
- The logarithmically interpolated value taken from the values for the fatigue strength and the static strength, based on the number of load cycles, is used for coefficients ZL, ZV, ZR, ZW, ZX, YdrelT, YRrelT and YX..

End of Report lines: 725

A.1.2 KISSsoft Report - Second stage

KISSsoft Release 03/2017 A

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File

Name : Unnamed

Changed by: Joana Mêda de Sousa on: 24.09.2017 at: 05:12:43

Important hint: At least one warning has occurred during the calculation:

1-> Special load spectrum:
Elements with power = 0 or speed = 0 are not usual!
(Element no. 4)

2-> Notice to gear 2:
NOT POSSIBLE TO MEASURE BASE TANGENT LENGTH!
The width of the gear is too small, hence the tooth thickness too big,
so that the required length for the measurement exceed the face width.

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:
Gear 1: IMGSUPROT(LSIMSUPPGCT)
Gear 2: LSGGEN(LSIMSUPPGCT)

Load spectrum

Own Input
Number of bins in the load spectrum: 11
Reference gear: 2

Bin	Frequency	Power	Speed	Torque	Coefficients						Temperature
No.	[%]	[kW]	[1/min]	[Nm]	KV	KHβ	KHα	Kγ	YM1	YM2	OilTemp
1	0.00124	-610.0002	77.9	-74791.0611	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
2	0.00103	-380.0001	77.9	-46591.1528	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
3	0.01066	-230.0001	77.9	-28199.9083	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
4	9.39362	0.0000	77.9	0.0000	1.0000	1.0000	0.0000	1.0000	1.0000	1.0000	0
5	27.24773	230.0001	77.9	28199.9083	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
6	19.30359	450.0002	77.9	55173.7336	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
7	13.09509	680.0002	77.9	83373.6419	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
8	23.13569	980.0004	77.9	120156.1310	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
9	7.79235	1210.0004	77.9	148356.0393	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
10	0.01894	1440.0005	77.9	176555.9475	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
11	0.00006	1670.0006	77.9	204755.8558	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65

Notice:
- Tooth flank with load spectrum: Check both cases and document the unfavorable case
- Tooth root with load spectrum: Check both cases and document the more realistic case (DIN3990-6, Method C)
  Is only applied on load spectrum bins, where the alternating bending factor (mean stress influence factor) YM=1.0.

S-N curve (Woehler line) in the endurance domain according: according to standard

Results

Safeties, calculated with load spectrum:

Root safety	1.696	1.560
Flank safety	1.287	1.286

Safeties against scuffing/micropitting/EHT/TFF are indicated for the most critical element of the load spectrum:

Scuffing safety (integral temperature)	4.177
Scuffing safety (flash temperature)	1.918
Safety against micropitting (B)	1.071

Safeties, calculated with nominal torque:

Safety against micropitting (B)	1.410

Analysis of critical elements in load spectrum: See section 10

ONLY AS INFORMATION: CALCULATION WITH REFERENCE POWER

Calculation method DIN 3990:1987 Method B

		------- GEAR 1	-------- GEAR 2 --
Power (kW)	[P]	1000.000	
Speed (1/min)	[n]	373.8	77.9
Torque (Nm)	[T]	25543.4	122608.3
Application factor	[KA]	1.00	
Required service life (h)	[H]	175200.00	
Gear driving (+) / driven (-)		-	+

Working flank gear 1: Right flank

Sense of rotation gear 1 counterclockwise

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

		------- GEAR 1	-------- GEAR 2 --
Center distance (mm)	[a]	642.500	
Centre distance tolerance	ISO 286:2010 Measure js7		
Normal module (mm)	[mn]	8.0000	
Pressure angle at normal section (°)	[alfn]	20.0000	
Helix angle at reference circle (°)	[beta]	25.0000	
Number of teeth	[z]	25	120
Double helical gearing		right/left	left/right
Total facewidth of Gear (mm)	[b]	188.00	188.00
Width of intermediate groove (mm)	[bNut]	40.00	
Facewidth for calculation (mm)	[beH]	74.00	74.00
Accuracy grade	[Q-DIN 3961:1978]	6	6
Inner diameter (mm)	[di]	0.00	0.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00

Material

Gear 1: 18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC

Gear 2: 18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC

		------- GEAR 1 --------	GEAR 2 --
Surface hardness		HRC 61	HRC 61
Fatigue strength. tooth root stress (N/mm²)	[σFlim]	500.00	500.00
Fatigue strength for Hertzian pressure (N/mm²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm²)	[σB]	1200.00	1200.00
Yield point (N/mm²)	[σS]	850.00	850.00
Young's modulus (N/mm²)	[E]	206000	206000
Poisson's ratio	[v]	0.300	0.300
Roughness average value DS, flank (µm)	[RAH]	0.60	0.60
Roughness average value DS, root (µm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (µm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (µm)	[RZF]	20.00	20.00

Gear reference profile 1 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000

not topping

Gear reference profile 2 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000

not topping

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofP*]	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification:for high load capacity gearboxe

Tip relief (µm)	[Ca]	80.0	80.0

Lubrication type		Oil injection lubrication
Type of oil (Own input)		Oil: Castrol Optigear Synthetic X 320
Lubricant base		Synthetic oil based on Polyalphaolefin
Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	325.00

Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	34.90	
Specific density at 15 °C (kg/dm³)	[roOil]	0.854	
Oil temperature (°C)	[TS]	65.000	

		------- GEAR 1 --------	GEAR 2 --
Overall transmission ratio	[itot]	-0.208	
Gear ratio	[u]	4.800	
Transverse module (mm)	[mt]	8.827	
Pressure angle at pitch circle (°)	[alft]	21.880	
Working transverse pressure angle (°)	[alfwt]	22.438	
	[alfwt.e/i]	22.446 / 22.429	
Working pressure angle at normal section (°)	[alfwn]	20.505	
Helix angle at operating pitch circle (°)	[betaw]	25.087	
Base helix angle (°)	[betab]	23.399	
Reference centre distance (mm)	[ad]	639.959	
Sum of profile shift coefficients	[Summexi]	0.3215	
Profile shift coefficient	[x]	0.2974	0.0241
Tooth thickness (Arc) (module) (module)	[sn*]	1.7873	1.5883
Tip alteration (mm)	[k*mn]	-0.031	-0.031
Reference diameter (mm)	[d]	220.676	1059.243
Base diameter (mm)	[db]	204.779	982.940
Tip diameter (mm)	[da]	241.371	1075.566
(mm)	[da.e/i]	241.371 / 241.361	1075.566 / 1075.556
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010
Tip form diameter (mm)	[dFa]	241.371	1075.566
(mm)	[dFa.e/i]	241.371 / 241.361	1075.566 / 1075.556
Active tip diameter (mm)	[dNa]	241.371	1075.566
Active tip diameter (mm)	[dNa.e/i]	241.371 / 241.361	1075.566 / 1075.556
Operating pitch diameter (mm)	[dw]	221.552	1063.448
(mm)	[dw.e/i]	221.566 / 221.538	1063.514 / 1063.382
Root diameter (mm)	[df]	205.433	1039.628
Generating Profile shift coefficient	[xE.e/i]	0.2810/ 0.2725	-0.0171/ -0.0343
Manufactured root diameter with xE (mm)	[df.e/i]	205.172 / 205.035	1038.969 / 1038.694
Theoretical tip clearance (mm)	[c]	2.000	2.000
Effective tip clearance (mm)	[c.e/i]	2.512 / 2.290	2.244 / 2.091
Active root diameter (mm)	[dNf]	211.728	1047.717
(mm)	[dNf.e/i]	211.787 / 211.675	1047.796 / 1047.645
Root form diameter (mm)	[dFf]	211.296	1044.353
(mm)	[dFf.e/i]	211.125 / 211.035	1043.756 / 1043.509
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.376 / 0.275	2.144 / 1.944
Addendum (mm)	[ha=mn*(haP*+x+k)]	10.348	8.162
(mm)	[ha.e/i]	10.348 / 10.343	8.162 / 8.157
Dedendum (mm)	[hf=mn*(hfP*-x)]	7.621	9.807
(mm)	[hf.e/i]	7.752 / 7.820	10.137 / 10.274
Roll angle at dFa (°)	[xsi_dFa.e/i]	35.750 / 35.744	25.453 / 25.452
Roll angle to dNa (°)	[xsi_dNa.e/i]	35.750 / 35.744	25.453 / 25.452
Roll angle to dNf (°)	[xsi_dNf.e/i]	15.118 / 14.993	21.154 / 21.129
Roll angle at dFf (°)	[xsi_dFf.e/i]	14.374 / 14.271	20.464 / 20.421
Tooth height (mm)	[h]	17.969	17.969
Virtual gear no. of teeth	[zn]	32.749	157.197
Normal tooth thickness at tip circle (mm)	[san]	5.397	6.576
(mm)	[san.e/i]	5.300 / 5.241	6.337 / 6.232
Normal-tooth thickness on tip form circle (mm)	[sFan]	5.397	6.576
(mm)	[sFan.e/i]	5.300 / 5.241	6.337 / 6.232
Normal space width at root circle (mm)	[efn]	6.526	5.635
(mm)	[efn.e/i]	6.591 / 6.627	5.662 / 5.674

Max. sliding velocity at tip (m/s)	[vga]	0.728	1.022
Specific sliding at the tip	[zetaa]	0.409	0.409
Specific sliding at the root	[zetaf]	-0.691	-0.691
Mean specific sliding	[zetam]	0.409	
Sliding factor on tip	[Kga]	0.236	0.168
Sliding factor on root	[Kgf]	-0.168	-0.236
Pitch on reference circle (mm)	[pt]	27.731	
Base pitch (mm)	[pbt]	25.733	
Transverse pitch on contact-path (mm)	[pet]	25.733	
Lead height (mm)	[pz]	1486.728	7136.296
Axial pitch (mm)	[px]	59.469	
Length of path of contact (mm)	[ga, e/i]	36.987 (37.092 / 36.861)	
Length T1-A, T2-A (mm)	[T1A, T2A]	63.886(63.886/ 63.876)	181.343(181.238/ 181.457)
Length T1-B (mm)	[T1B, T2B]	52.632(52.527/ 52.749)	192.597(192.597/ 192.584)
Length T1-C (mm)	[T1C, T2C]	42.281(42.263/ 42.299)	202.948(202.861/ 203.034)
Length T1-D (mm)	[T1D, T2D]	38.153(38.153/ 38.143)	207.076(206.971/ 207.190)
Length T1-E (mm)	[T1E, T2E]	26.898(26.794/ 27.015)	218.330(218.330/ 218.318)
Length T1-T2 (mm)	[T1T2]	245.228 (245.124 / 245.333)	
Diameter of single contact point B (mm)	[d-B]	230.250(218.534/ 218.527)	1055.720(1066.545/1066.716)
Diameter of single contact point D (mm)	[d-D]	218.534(230.154/ 230.357)	1066.627(1055.720/1055.711)
Addendum contact ratio	[eps]	0.840(0.840/ 0.839)	0.598(0.601/ 0.594)
Minimal length of contact line (mm)	[Lmin]	108.970	

Transverse contact ratio	[eps_a]	1.437
Transverse contact ratio with allowances	[eps_a.e/m/i]	1.441 / 1.437 / 1.432
Overlap ratio	[eps_b]	1.244
Total contact ratio	[eps_g]	2.682
Total contact ratio with allowances	[eps_g.e/m/i]	2.686 / 2.681 / 2.677

2. FACTORS OF GENERAL INFLUENCE

		------- GEAR 1 -------- GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]	231501.8
Axial force (N)	[Fa]	0.0
Radial force (N)	[Fr]	92970.4
Normal force (N)	[Fnorm]	271827.1
Nominal circumferential force per mm (N/mm)	[w]	1564.20
Only as information: Forces at operating pitch circle:		
Nominal circumferential force (N)	[Ftw]	230586.3
Axial force (N)	[Faw]	0.0
Radial force (N)	[Frw]	95218.3
Circumferential speed reference circle (m/s)	[v]	4.32
Circumferential speed operating pitch circle (m/s)	[v(dw)]	4.34
Running-in value (µm)	[yp]	1.0
Running-in value (µm)	[yf]	1.1
Correction coefficient	[CM]	0.800
Gear body coefficient	[CR]	1.000
Basic rack factor	[CBS]	0.975
Material coefficient	[E/Est]	1.000
Singular tooth stiffness (N/mm/µm)	[c']	13.845
Meshing stiffness (N/mm/µm)	[cg]	18.387
Reduced mass (kg/mm)	[mRed]	0.17463
Resonance speed (min-1)	[nE1]	3919

User specified factor KV:

Dynamic factor	[KV]	1.050	
User specified factor KHb:			
Face load factor - flank	[KHb]	1.150	
- Tooth root	[KFb]	1.113	
- Scuffing	[KBb]	1.150	
User specified factor KHa:			
Transverse load factor - flank	[KHa]	1.000	
- Tooth root	[KFa]	1.000	
- Scuffing	[KBa]	1.000	
Helical load factor scuffing	[Kbg]	1.251	
Number of load cycles (in mio.)	[NL]	3929.871	1637.446

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B

		------- GEAR 1	-------- GEAR 2 --
Calculated with profile shift	[x]	0.2974	0.0241
Tooth form factor	[YF]	1.09	1.20
Stress correction factor	[YS]	2.30	2.27
Working angle (°)	[alfFen]	20.66	20.03
Bending moment arm (mm)	[hF]	7.04	8.42
Tooth thickness at root (mm)	[sFn]	17.58	18.35
Tooth root radius (mm)	[roF]	3.67	3.51

(hF* = 0.881/ 1.053 sFn* = 2.198/ 2.294 roF* = 0.459/ 0.439)
(den (mm) =
267.873/1261.821 dsFn(mm) =208.458/1042.793 alfsFn(°) = 30.00/ 30.00 qs = 2.393/ 2.615)

Contact ratio factor	[Yeps]	1.000	
Helix angle factor	[Ybet]	0.792	
Effective facewidth (mm)	[beff]	74.00	74.00
Nominal stress at tooth root (N/mm²)	[sigF0]	386.90	421.27
Tooth root stress (N/mm²)	[sigF]	451.99	492.15
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdrelT]	0.999	1.001
Surface factor	[YRrelT]	0.957	0.957
size factor (Tooth root)	[YX]	0.970	0.970
Finite life factor	[YNT]	1.000	1.000
	[YdrelT*YRrelT*YX*YNT]	0.927	0.929
Alternating bending factor (mean stress influence coefficient)	[YM]	1.000	1.000
Stress correction factor	[Yst]	2.00	
Yst*sigFlim (N/mm²)	[sigFE]	1000.00	1000.00
Permissible tooth root stress (N/mm²)	[sigFP=sigFG/SFmin]	594.29	595.54
Limit strength tooth root (N/mm²)	[sigFG]	927.09	929.04
Required safety	[SFmin]	1.56	1.56

4. SAFETY AGAINST PITTING (TOOTH FLANK)

------- GEAR 1 -------- GEAR 2 --

Zone factor	[ZH]	2.272	
Elasticity factor (√N/mm²)	[ZE]	189.812	
Contact ratio factor	[Zeps]	0.834	
Helix angle factor	[Zbet]	0.952	
Effective facewidth (mm)	[beff]	74.00	
Nominal contact stress (N/mm²)	[sigH0]	1002.19	
Contact stress at operating pitch circle (N/mm²)	[sigHw]	1101.27	
Single tooth contact factor	[ZB,ZD]	1.00	1.00
Contact stress (N/mm²)	[sigHB, sigHD]	1101.27	1101.27
Lubrication coefficient at NL	[ZL]	1.048	1.048
Speed coefficient at NL	[ZV]	0.979	0.979
Roughness coefficient at NL	[ZR]	1.012	1.012
Material pairing coefficient at NL	[ZW]	1.000	1.000
Finite life factor	[ZNT]	1.000	1.000
	[ZL*ZV*ZR*ZNT]	1.039	1.039
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress (N/mm²)	[sigHP=sigHG/SHmin]	1246.47	1246.47
Pitting stress limit (N/mm²)	[sigHG]	1558.08	1558.08
Required safety	[SHmin]	1.25	1.25

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Calculation of permissible specific film thickness

Lubricant load according to FVA Info sheet 54/7 10 (Oil: Castrol Optigear Synthetic X 320)

Reference data FZG-C Test:

Torque (Nm)	[T1Ref]	265.1
Line load at contact point A (N/mm)	[FbbRef,A]	236.3
Oil temperature (°C)	[theOilRef]	60.0
Tooth mass temperature (°C)	[theMRef]	90.3
Contact temperature (°C)	[theBRef,A]	181.8
Lubrication gap thickness (µm)	[hRef,A]	0.130
Specific film thickness in test (µm)	[lamGFT]	0.260
Material coefficient	[WW]	1.00
Permissible specific film thickness (µm)	[lamGFP]	0.363

Interim results in accordance with ISO/TR 15144:2014

Coefficient of friction	[mym]	0.041
Lubricant factor	[XL]	0.800
Roughness factor	[XR]	0.779
Tooth mass temperature (°C)	[theM]	67.7
Tip relief factor	[XCa (A)]	1.702
Loss factor	[HV]	0.103
Equivalent Young's modulus (N/mm²)	[Er]	226374
Pressure-viscosity coefficient (m²/N)	[alf38]	0.01380
Dynamic viscosity (Ns/m²)	[etatM]	79.4
Roughness average value (µm)	[Ra]	0.6

Calculation of speeds, load distribution and flank curvature according to method B following ISO/TR 15144-1:2014

Ca taken as optimal in the calculation (0=no, 1=yes) 1 1

5. SCUFFING LOAD CAPACITY

Calculation method according to DIN 3990:1987

Lubrication coefficient (for lubrication type)	[XS]	1.200		
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)		14
Relative structure coefficient (Scuffing)	[XWrelT]	1.000		
Thermal contact factor (N/mm/s^.5/K)	[BM]	13.780	13.780	
Relevant tip relief (µm)	[Ca]	80.00	80.00	
Optimal tip relief (µm)	[Ceff]	66.97		
Ca taken as optimal in the calculation (0=no, 1=yes)		1	1	
Effective facewidth (mm)	[beff]	74.000		
Applicable circumferential force/facewidth (N/mm)	[wBt]	2363.656		
Angle factor	[Xalfbet]	0.995		
(ε1:0.840, ε2:0.598)				

Flash temperature-criteria

Tooth mass temperature (°C)	[theMB]	106.00
(theMB = theoil + XS*0.47*theflamax)		
Maximum flash temperature (°C)	[theflamax]	72.69
Scuffing temperature (°C)	[theS]	528.15
Coordinate gamma (point of highest temp.)	[Gamma]	0.245
[Gamma.A]=0.511 [Gamma.E]=-0.364		
Highest contact temp. (°C)	[theB]	178.69
Flash factor (°K*N^-.75*s^.5*m^-.5*mm)	[XM]	50.058
Geometry factor	[XB]	0.113
Load sharing factor	[XGam]	1.000
Dynamic viscosity (mPa*s)	[etaM]	24.55 (65.0 °C)
Coefficient of friction	[mym]	0.093

Integral temperature-criteria

Tooth mass temperature (°C)	[theMC]	81.22
(theMC = theoil + XS*0.70*theflaint)		
Mean flash temperature (°C)	[theflaint]	19.31
Integral scuffing temperature (°C)	[theSint]	528.15
Flash factor (°K*N^-.75*s^.5*m^-.5*mm)	[XM]	50.058
Contact ratio factor	[Xeps]	0.268
Dynamic viscosity (mPa*s)	[etaOil]	88.12 (65.0 °C)
Mean coefficient of friction	[mym]	0.072
Geometry factor	[XBE]	0.219
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.516
Integral tooth flank temperature (°C)	[theint]	110.19

6. MEASUREMENTS FOR TOOTH THICKNESS

		------- Gear 1 ------	Gear 2 --
Tooth thickness deviation		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.095 / -0.145	-0.240 / -0.340
Number of teeth spanned	[k]	5.000	18.000
Base tangent length (no backlash) (mm)	[Wk]	111.609	431.215
Actual base tangent length ('span') (mm)	[Wk.e/i]	111.520 / 111.473	430.989 / 430.895
(mm)	[ΔWk.e/i]	-0.089 / -0.136	-0.226 / -0.319
Diameter of contact point (mm)	[dMWk.m]	228.922	1059.525
> Base tangent length Gear 2 is not measurable (Gear to thin)			
Theoretical diameter of ball/pin (mm)	[DM]	14.234	13.428
Effective diameter of ball/pin (mm)	[DMeff]	15.000	14.000
Radial single-ball measurement backlash free (mm)	[MrK]	123.969	539.954

Radial single-ball measurement (mm)	[MrK.e/i]	123.867 / 123.813	539.636 / 539.503
Diameter of contact point (mm)	[dMMr.m]	226.468	1060.256
Diametral measurement over two balls without clearance (mm)	[MdK]	247.479	1079.908
Diametral two ball measure (mm)	[MdK.e/i]	247.275 / 247.167	1079.272 / 1079.006
Diametral measurement over pins without clearance (mm)	[MdR]	247.939	1079.908
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	247.734 / 247.626	1079.272 / 1079.006
Measurement over 2 pins (free) according to AGMA 2002 (mm)	[dk2f.e/i]	247.129 / 247.022	0.000 / 0.000
Measurement over 2 pins (transverse) according to AGMA 2002 (mm)	[dk2t.e/i]	248.182 / 248.074	0.000 / 0.000
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	247.734 / 247.626	1079.272 / 1079.006
Chordal tooth thickness (no backlash) (mm)	[sc]	14.291	12.707
Actual chordal tooth thickness (mm)	[sc.e/i]	14.196 / 14.146	12.467 / 12.367
Reference chordal height from da.m (mm)	[ha]	10.536	8.191
Tooth thickness (Arc) (mm)	[sn]	14.298	12.707
(mm)	[sn.e/i]	14.203 / 14.153	12.467 / 12.367
Backlash free center distance (mm)	[aControl.e/i]	642.050 /641.848	
Backlash free center distance, allowances (mm)	[jta]	-0.450 / -0.652	
dNf.i with aControl (mm)	[dNf0.i]	210.882	1046.535
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	-0.121	1.389
Tip clearance (mm)	[c0.i(aControl)]	1.677	1.478
Centre distance allowances (mm)	[Aa.e/i]	0.040 / -0.040	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.033 / -0.033	
Radial clearance (mm)	[jrw]	0.692 / 0.410	
Circumferential backlash (transverse section) (mm)	[jtw]	0.570 / 0.338	
Normal backlash (mm)	[jnw]	0.486 / 0.288	
Torsional angle at entry with fixed output:			
Entire torsional angle (°)	[j.tSys]	0.0615/ 0.0364	

7. GEAR ACCURACY

		------- GEAR 1	-------- GEAR 2 --
According to DIN 3961:1978			
Accuracy grade	[Q-DIN3961]	6	6
Profile form deviation (µm)	[ff]	14.00	14.00
Profile slope deviation (µm)	[fHa]	9.00	9.00
Total profile deviation (µm)	[Ff]	16.00	16.00
Helix form deviation (µm)	[fbf]	8.00	8.00
Helix slope deviation (µm)	[fHb]	10.00	10.00
Total helix deviation (µm)	[Fb]	13.00	13.00
Normal base pitch deviation (µm)	[fpe]	11.00	13.00
Single pitch deviation (µm)	[fp]	11.00	13.00
Adjacent pitch difference (µm)	[fu]	13.00	16.00
Total cumulative pitch deviation (µm)	[Fp]	37.00	54.00
Sector pitch deviation over z/8 pitches (µm)	[Fpz/8]	24.00	34.00
Runout (µm)	[Fr]	28.00	39.00
Tooth Thickness Variation (µm)	[Rs]	16.00	23.00
Single flank composite, total (µm)	[Fi']	43.00	56.00
Single flank composite, tooth-to-tooth (µm)	[fi']	19.00	21.00
Radial composite, total (µm)	[Fi"]	31.00	42.00
Radial composite, tooth-to-tooth (µm)	[fi"]	14.00	19.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality) 6)

Maximum value for deviation error of axis (µm)	[fSigbet]	36.58 (Fb=18.00)
Maximum value for inclination error of axes (µm)	[fSigdel]	73.17

8. ADDITIONAL DATA

Mass (kg)	[m]	57.701	1293.154
Total mass (kg)	[m]	1350.855	
Moment of inertia (system with reference to the drive):			
calculation without consideration of the exact tooth shape			
single gears ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.35997	180.80111
System ((da+df)/2...di) (kg*m²)	[TraeghMom]	189.09490	
Torsional stiffness at entry with driven force fixed:			
Torsional stiffness (MNm/rad)	[cr]	654.698	
Torsion when subjected to nominal torque (°)	[delcr]	0.011	
Mean coeff. of friction (acc. Niemann)	[mum]	0.042	
Wear sliding coef. by Niemann	[zetw]	0.587	
Gear power loss (kW)	[PVZ]	4.339	
(Meshing efficiency (%)	[etaz]	99.566)	
Sound pressure level (according to Masuda)	[dB(A)]	100.7	
Oil requirement for injection lubrication (l/min)			
	[Voil]	13.665	

(with oil cooler, for assumed difference in temperature of oil (°C): 10)

Classification according to F.E.M. (Edition 1.001, 1998)

Spectrum factor	[km]	0.090
Spectrum class	[L]	1
Application class (predefined service life)	[T]	9
Machine class (predefined service life)	[M]	8
Application class (achievable service life)	[T]	9
Machine class (achievable service life)	[M]	8

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Profile and tooth trace modifications for gear 1

Symmetric (both flanks)

- Tip relief, linear Caa = 80.000µm LCa = 0.717*mn dCa = 235.498mm

Profile and tooth trace modifications for gear 2

Symmetric (both flanks)

- Tip relief, linear Caa = 80.000µm LCa = 0.717*mn dCa =1070.959mm

Tip relief verification

Diameter (mm)	[dcheck]	241.201		1075.396	
Tip relief left/right (µm)	[Ca L/R]	77.8 /	77.8	77.1 /	77.1

Data for the tooth form calculation :
Data not available.

10. SERVICE LIFE, DAMAGE

Calculation with load spectrum

Required safety for tooth root	[SFmin]	1.56
Required safety for tooth flank	[SHmin]	1.25

Service life (calculated with required safeties):

System service life (h)	[Hatt]	364288	
Tooth root service life (h)	[HFatt]	3.643e+005	4.742e+005
Tooth flank service life (h)	[HHatt]	1e+006	1e+006

Note: The entry 1e+006 h means that the Service life > 1,000,000 h.

Damage calculated on the basis of the required service life [H] (175200.0 h)

No.	F1%	F2%	H1%	H2%
1	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00
9	0.00	0.00	0.00	0.00
10	47.69	36.63	3.45	1.44
11	0.40	0.31	0.03	0.01
Σ	48.09	36.94	3.48	1.45

Damage calculated on basis of system service life [Hatt] (364288.0 h)

No.	F1%	F2%	H1%	H2%
1	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00
9	0.00	0.00	0.00	0.00
10	99.17	76.17	7.18	2.99
11	0.83	0.65	0.06	0.03
Σ	100.00	76.82	7.24	3.02

Damage calculated on basis of individual service life HFatt & HHatt

	HFatt1	HFatt2	HHatt1	HHatt2
(h)	3.643e+005	4.742e+005	1e+006	1e+006
No.	F1%	F2%	H1%	H2%
1	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00
9	0.00	0.00	0.00	0.00
10	99.17	99.15	99.16	99.16

11   0.83   0.85   0.84   0.84

Σ   100.00   100.00   100.00   100.00

Most critical duty cycle elements for Scoring (SB, Sint), Tooth Flank Fracture (SFF), hardened layer (SEHT) and Micropitting (Slam)

Calculation of the factors required to define reliability R(t) according to B. Bertsche with Weibull distribution:
R(t) = 100 * Exp(-((t*fac - t0)/(T - t0))^b) %; t (h)

Gear		fac	b	t0	T	R(H)%
1	Tooth root	22431	1.7	7.888e+009	1.212e+010	100.00
1	Tooth flank	22431	1.3	1.017e+011	4.848e+011	100.00
2	Tooth root	9346	1.7	4.279e+009	6.576e+009	100.00
2	Tooth flank	9346	1.3	1.017e+011	4.848e+011	100.00

Reliability of the configuration for required service life (%)   100.00   (Bertsche)

REMARKS:

- Specifications with   [.e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances
  Specifications with   [.m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness
deviation are taken into account. Shown is the maximal and the minimal backlash corresponding
 the largest resp. the smallest allowances
 The calculation is done for the operating pitch circle.
- Details of calculation method:
  cg according to method B

End of Report   lines: 638

A.1.3 KISSsoft Report - Third stage

KISSsoft Release 03/2017 A

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File

Name : Unnamed
Changed by: Joana Mêda de Sousa on: 24.09.2017 at: 05:14:38

Important hint: At least one warning has occurred during the calculation:

1-> Special load spectrum:
Elements with power = 0 or speed = 0 are not usual!
(Element no. 4)

2-> Notice to gear 2:
NOT POSSIBLE TO MEASURE BASE TANGENT LENGTH!
The width of the gear is too small, hence the tooth thickness too big,
so that the required length for the measurement exceed the face width.

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:
Gear 1: HSG(IMHSSUPPGCT)
Gear 2: IMGSUPGEN(IMHSSUPPGCT)

Load spectrum

Own Input
Number of bins in the load spectrum: 11
Reference gear: 2

Bin No.	Frequency [%]	Power [kW]	Speed [1/min]	Torque [Nm]	Coefficients KV	KHβ	KHα	Kγ	YM1	YM2	Temperature OilTemp
1	0.00124	-610.0002	373.8	-15581.4710	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
2	0.00103	-380.0001	373.8	-9706.4901	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
3	0.01066	-230.0001	373.8	-5874.9809	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
4	9.39362	0.0000	373.8	0.0000	1.0000	1.0000	0.0000	1.0000	1.0000	1.0000	0
5	27.24773	230.0001	373.8	5874.9809	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
6	19.30359	450.0002	373.8	11494.5278	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
7	13.09509	680.0002	373.8	17369.5086	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
8	23.13569	980.0004	373.8	25032.5272	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
9	7.79235	1210.0004	373.8	30907.5080	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
10	0.01894	1440.0005	373.8	36782.4889	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65
11	0.00006	1670.0006	373.8	42657.4697	1.0500	1.1500	1.0000	1.0000	1.0000	1.0000	65

Notice:
- Tooth flank with load spectrum: Check both cases and document the unfavorable case
- Tooth root with load spectrum: Check both cases and document the more realistic case (DIN3990-6, Method C)
  Is only applied on load spectrum bins, where the alternating bending factor (mean stress influence factor) YM=1.0.

S-N curve (Woehler line) in the endurance domain according: according to standard

Results

Safeties, calculated with load spectrum:

Root safety	1.566	1.593
Flank safety	1.323	1.321

Safeties against scuffing/micropitting/EHT/TFF are indicated for the most critical element of the load spectrum:

Scuffing safety (integral temperature)	3.754
Scuffing safety (flash temperature)	1.995
Safety against micropitting (B)	1.500

Safeties, calculated with nominal torque:

Safety against micropitting (B)	2.074

Analysis of critical elements in load spectrum: See section 10

ONLY AS INFORMATION: CALCULATION WITH REFERENCE POWER

Calculation method DIN 3990:1987 Method B

		------- GEAR 1	-------- GEAR 2 --
Power (kW)	[P]	1000.000	
Speed (1/min)	[n]	1495.4	373.8
Torque (Nm)	[T]	6385.8	25543.4
Application factor	[KA]	1.00	
Required service life (h)	[H]	175200.00	
Gear driving (+) / driven (-)		-	+

Working flank gear 1: Left flank

Sense of rotation gear 1 clockwise

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

		------- GEAR 1	-------- GEAR 2 --
Center distance (mm)	[a]	346.500	
Centre distance tolerance	ISO 286:2010 Measure js7		
Normal module (mm)	[mn]	5.0000	
Pressure angle at normal section (°)	[alfn]	20.0000	
Helix angle at reference circle (°)	[beta]	25.0000	
Number of teeth	[z]	25	100
Double helical gearing		left/right	right/left
Total facewidth of Gear (mm)	[b]	127.00	127.00
Width of intermediate groove (mm)	[bNut]	25.00	
Facewidth for calculation (mm)	[beH]	51.00	51.00
Accuracy grade	[Q-DIN 3961:1978]	6	6
Inner diameter (mm)	[di]	0.00	0.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00

Material

Gear 1: 18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC

Gear 2: 18CrNiMo7-6, Case-carburized steel, case-hardened

ISO 6336-5 Figure 9/10 (MQ), Core hardness >=30HRC

		------- GEAR 1	-------- GEAR 2 --
Surface hardness		HRC 61	HRC 61
Fatigue strength. tooth root stress (N/mm²)	[σFlim]	500.00	500.00
Fatigue strength for Hertzian pressure (N/mm²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm²)	[σB]	1200.00	1200.00
Yield point (N/mm²)	[σS]	850.00	850.00
Young's modulus (N/mm²)	[E]	206000	206000
Poisson's ratio	[v]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.60	0.60
Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	4.80	4.80
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

Gear reference profile 1 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000

not topping

Gear reference profile 2 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000

not topping

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofP*]	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification:for high load capacity gearboxe

Tip relief (μm)	[Ca]	50.0	50.0

Lubrication type		Oil injection lubrication
Type of oil (Own input)		Oil: Castrol Optigear Synthetic X 320
Lubricant base		Synthetic oil based on Polyalphaolefin
Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	325.00

Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	34.90	
Specific density at 15 °C (kg/dm³)	[roOil]	0.854	
Oil temperature (°C)	[TS]	65.000	

		------- GEAR 1 --------	GEAR 2 --
Overall transmission ratio	[itot]	-0.250	
Gear ratio	[u]	4.000	
Transverse module (mm)	[mt]	5.517	
Pressure angle at pitch circle (°)	[alft]	21.880	
Working transverse pressure angle (°)	[alfwt]	22.568	
	[alfwt.e/i]	22.579 / 22.556	
Working pressure angle at normal section (°)	[alfwn]	20.623	
Helix angle at operating pitch circle (°)	[betaw]	25.108	
Base helix angle (°)	[betab]	23.399	
Reference centre distance (mm)	[ad]	344.806	
Sum of profile shift coefficients	[Summexi]	0.3440	
Profile shift coefficient	[x]	0.2964	0.0476
Tooth thickness (Arc) (module) (module)	[sn*]	1.7866	1.6054
Tip alteration (mm)	[k*mn]	-0.025	-0.025
Reference diameter (mm)	[d]	137.922	551.689
Base diameter (mm)	[db]	127.987	511.948
Tip diameter (mm)	[da]	150.836	562.115
(mm)	[da.e/i]	150.836 / 150.826	562.115 / 562.105
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010
Tip form diameter (mm)	[dFa]	150.836	562.115
(mm)	[dFa.e/i]	150.836 / 150.826	562.115 / 562.105
Active tip diameter (mm)	[dNa]	150.836	562.115
Active tip diameter (mm)	[dNa.e/i]	150.836 / 150.826	562.115 / 562.105
Operating pitch diameter (mm)	[dw]	138.600	554.400
(mm)	[dw.e/i]	138.611 / 138.589	554.446 / 554.354
Root diameter (mm)	[df]	128.386	539.665
Generating Profile shift coefficient	[xE.e/i]	0.2703/ 0.2566	0.0118/ -0.0046
Manufactured root diameter with xE (mm)	[df.e/i]	128.125 / 127.988	539.307 / 539.143
Theoretical tip clearance (mm)	[c]	1.250	1.250
Effective tip clearance (mm)	[c.e/i]	1.544 / 1.400	1.482 / 1.352
Active root diameter (mm)	[dNf]	132.383	544.737
(mm)	[dNf.e/i]	132.427 / 132.345	544.794 / 544.686
Root form diameter (mm)	[dFf]	132.054	542.683
(mm)	[dFf.e/i]	131.883 / 131.795	542.366 / 542.220
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.316 / 0.231	1.287 / 1.160
Addendum (mm)	[ha=mn*(haP*+x+k)]	6.457	5.213
(mm)	[ha.e/i]	6.457 / 6.452	5.213 / 5.208
Dedendum (mm)	[hf=mn*(hfP*-x)]	4.768	6.012
(mm)	[hf.e/i]	4.899 / 4.967	6.191 / 6.273
Roll angle at dFa (°)	[xsi_dFa.e/i]	35.732 / 35.724	25.979 / 25.976
Roll angle to dNa (°)	[xsi_dNa.e/i]	35.732 / 35.724	25.979 / 25.976
Roll angle to dNf (°)	[xsi_dNf.e/i]	15.222 / 15.078	20.851 / 20.815
Roll angle at dFf (°)	[xsi_dFf.e/i]	14.245 / 14.080	20.042 / 19.993
Tooth height (mm)	[h]	11.225	11.225
Virtual gear no. of teeth	[zn]	32.749	130.998
Normal tooth thickness at tip circle (mm)	[san]	3.380	4.084
(mm)	[san.e/i]	3.284 / 3.225	3.956 / 3.891
Normal-tooth thickness on tip form circle (mm)	[sFan]	3.380	4.084
(mm)	[sFan.e/i]	3.284 / 3.225	3.956 / 3.891
Normal space width at root circle (mm)	[efn]	4.081	3.557
(mm)	[efn.e/i]	4.148 / 4.187	3.575 / 3.583

Max. sliding velocity at tip (m/s)	[vga]	1.895	2.606
Specific sliding at the tip	[zetaa]	0.417	0.417
Specific sliding at the root	[zetaf]	-0.715	-0.715
Mean specific sliding	[zetam]	0.417	
Sliding factor on tip	[Kga]	0.240	0.175
Sliding factor on root	[Kgf]	-0.175	-0.240
Pitch on reference circle (mm)	[pt]	17.332	
Base pitch (mm)	[pbt]	16.083	
Transverse pitch on contact-path (mm)	[pet]	16.083	
Lead height (mm)	[pz]	929.205	3716.821
Axial pitch (mm)	[px]	37.168	
Length of path of contact (mm)	[ga, e/i]	22.994 (23.068 / 22.898)	
Length T1-A, T2-A (mm)	[T1A, T2A]	39.909(39.909/ 39.900)	93.069(92.994/ 93.152)
Length T1-B (mm)	[T1B, T2B]	32.998(32.924/ 33.085)	99.979(99.979/ 99.967)
Length T1-C (mm)	[T1C, T2C]	26.596(26.581/ 26.610)	106.382(106.323/ 106.442)
Length T1-D (mm)	[T1D, T2D]	23.826(23.826/ 23.816)	109.152(109.078/ 109.236)
Length T1-E (mm)	[T1E, T2E]	16.915(16.841/ 17.001)	116.063(116.063/ 116.051)
Length T1-T2 (mm)	[T1T2]	132.978 (132.903 / 133.052)	
Diameter of single contact point B (mm)	[d-B]	144.001(136.570/ 136.563)	549.613(556.491/ 556.615)
Diameter of single contact point D (mm)	[d-D]	136.570(143.933/ 144.080)	556.550(549.613/ 549.604)
Addendum contact ratio	[eps]	0.828(0.829/ 0.826)	0.602(0.606/ 0.597)
Minimal length of contact line (mm)	[Lmin]	72.971	

Transverse contact ratio	[eps_a]	1.430
Transverse contact ratio with allowances	[eps_a.e/m/i]	1.434 / 1.429 / 1.424
Overlap ratio	[eps_b]	1.372
Total contact ratio	[eps_g]	2.802
Total contact ratio with allowances	[eps_g.e/m/i]	2.806 / 2.801 / 2.796

2. FACTORS OF GENERAL INFLUENCE

		------- GEAR 1 -------- GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]	92600.7
Axial force (N)	[Fa]	0.0
Radial force (N)	[Fr]	37188.1
Normal force (N)	[Fnorm]	108730.9
Nominal circumferential force per mm (N/mm)	[w]	907.85
Only as information: Forces at operating pitch circle:		
Nominal circumferential force (N)	[Ftw]	92147.9
Axial force (N)	[Faw]	0.0
Radial force (N)	[Frw]	38296.4
Circumferential speed reference circle (m/s)	[v]	10.80
Circumferential speed operating pitch circle (m/s)	[v(dw)]	10.85

Running-in value (µm)	[yp]	0.8
Running-in value (µm)	[yf]	0.8
Correction coefficient	[CM]	0.800
Gear body coefficient	[CR]	1.000
Basic rack factor	[CBS]	0.975
Material coefficient	[E/Est]	1.000
Singular tooth stiffness (N/mm/µm)	[c']	13.780
Meshing stiffness (N/mm/µm)	[cg]	18.221
Reduced mass (kg/mm)	[mRed]	0.06690
Resonance speed (min-1)	[nE1]	6304

User specified factor KV:

Dynamic factor	[KV]	1.050	
User specified factor KHb:			
Face load factor - flank	[KHb]	1.150	
- Tooth root	[KFb]	1.116	
- Scuffing	[KBb]	1.150	
User specified factor KHa:			
Transverse load factor - flank	[KHa]	1.000	
- Tooth root	[KFa]	1.000	
- Scuffing	[KBa]	1.000	
Helical load factor scuffing	[Kbg]	1.266	
Number of load cycles (in mio.)	[NL]	15719.483	7859.742

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B

		------- GEAR 1 --------	GEAR 2 --
Calculated with profile shift	[x]	0.2964	0.0476
Tooth form factor	[YF]	1.10	1.21
Stress correction factor	[YS]	2.28	2.25
Working angle (°)	[alfFen]	20.75	20.09
Bending moment arm (mm)	[hF]	4.45	5.26
Tooth thickness at root (mm)	[sFn]	10.99	11.43
Tooth root radius (mm)	[roF]	2.30	2.22

(hF* = 0.891/ 1.052 sFn* = 2.198/ 2.285 roF* = 0.460/ 0.444)
(den (mm) =
167.510/657.867 dsFn(mm) = 130.277/541.647 alfsFn(°) = 30.00/ 30.00 qs = 2.391/ 2.573)

Contact ratio factor	[Yeps]	1.000	
Helix angle factor	[Ybet]	0.792	
Effective facewidth (mm)	[beff]	51.00	51.00
Nominal stress at tooth root (N/mm²)	[sigF0]	361.58	390.47
Tooth root stress (N/mm²)	[sigF]	423.66	457.50
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdrelT]	0.999	1.001
Surface factor	[YRrelT]	0.957	0.957
size factor (Tooth root)	[YX]	1.000	1.000
Finite life factor	[YNT]	1.000	1.000
	[YdrelT*YRrelT*YX*YNT]	0.956	0.957
Alternating bending factor (mean stress influence coefficient)	[YM]	1.000	1.000
Stress correction factor	[Yst]	2.00	
Yst*sigFlim (N/mm²)	[sigFE]	1000.00	1000.00
Permissible tooth root stress (N/mm²)	[sigFP=sigFG/SFmin]	612.66	613.71
Limit strength tooth root (N/mm²)	[sigFG]	955.75	957.39
Required safety	[SFmin]	1.56	1.56

4. SAFETY AGAINST PITTING (TOOTH FLANK)

------- GEAR 1 -------- GEAR 2 --

Zone factor	[ZH]	2.265	
Elasticity factor (√N/mm²)	[ZE]	189.812	
Contact ratio factor	[Zeps]	0.836	
Helix angle factor	[Zbet]	0.952	
Effective facewidth (mm)	[beff]	51.00	
Nominal contact stress (N/mm²)	[sigH0]	981.75	
Contact stress at operating pitch circle (N/mm²)	[sigHw]	1078.80	
Single tooth contact factor	[ZB,ZD]	1.00	1.00
Contact stress (N/mm²)	[sigHB, sigHD]	1078.80	1078.80
Lubrication coefficient at NL	[ZL]	1.048	1.048
Speed coefficient at NL	[ZV]	1.002	1.002
Roughness coefficient at NL	[ZR]	0.996	0.996
Material pairing coefficient at NL	[ZW]	1.000	1.000
Finite life factor	[ZNT]	1.000	1.000
	[ZL*ZV*ZR*ZNT]	1.046	1.046
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress (N/mm²)	[sigHP=sigHG/SHmin]	1255.30	1255.30
Pitting stress limit (N/mm²)	[sigHG]	1569.13	1569.13
Required safety	[SHmin]	1.25	1.25

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Calculation of permissible specific film thickness

Lubricant load according to FVA Info sheet 54/7 10 (Oil: Castrol Optigear Synthetic X 320)

Reference data FZG-C Test:

Torque (Nm)	[T1Ref]	265.1
Line load at contact point A (N/mm)	[FbbRef,A]	236.3
Oil temperature (°C)	[theOilRef]	60.0
Tooth mass temperature (°C)	[theMRef]	90.3
Contact temperature (°C)	[theBRef,A]	181.8
Lubrication gap thickness (µm)	[hRef,A]	0.130
Specific film thickness in test (µm)	[lamGFT]	0.260
Material coefficient	[WW]	1.00
Permissible specific film thickness (µm)	[lamGFP]	0.363

Interim results in accordance with ISO/TR 15144:2014

Coefficient of friction	[mym]	0.038
Lubricant factor	[XL]	0.800
Roughness factor	[XR]	0.882
Tooth mass temperature (°C)	[theM]	70.4
Tip relief factor	[XCa (A)]	1.685
Loss factor	[HV]	0.106
Equivalent Young's modulus (N/mm²)	[Er]	226374
Pressure-viscosity coefficient (m²/N)	[alf38]	0.01380
Dynamic viscosity (Ns/m²)	[etatM]	71.6
Roughness average value (µm)	[Ra]	0.6

Calculation of speeds, load distribution and flank curvature according to method B following ISO/TR 15144-1:2014

Ca taken as optimal in the calculation (0=no, 1=yes) 1 1

5. SCUFFING LOAD CAPACITY

Calculation method according to DIN 3990:1987

Lubrication coefficient (for lubrication type)	[XS]	1.200	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	14
Relative structure coefficient (Scuffing)	[XWrelT]	1.000	
Thermal contact factor (N/mm/s^.5/K)	[BM]	13.780	13.780
Relevant tip relief (µm)	[Ca]	50.00	50.00
Optimal tip relief (µm)	[Ceff]	40.02	
Ca taken as optimal in the calculation (0=no, 1=yes)		1	1
Effective facewidth (mm)	[beff]	51.000	
Applicable circumferential force/facewidth (N/mm)	[wBt]	1387.301	
Angle factor	[Xalfbet]	0.996	
(ε1:0.828, ε2:0.602)			

Flash temperature-criteria

Tooth mass temperature (°C)	[theMB]	104.65
(theMB = theoil + XS*0.47*theflamax)		
Maximum flash temperature (°C)	[theflamax]	70.30
Scuffing temperature (°C)	[theS]	528.15
Coordinate gamma (point of highest temp.)	[Gamma]	0.241
[Gamma.A]=0.501 [Gamma.E]=-0.364		
Highest contact temp. (°C)	[theB]	174.95
Flash factor (°K*N^-.75*s^.5*m^-.5*mm)	[XM]	50.058
Geometry factor	[XB]	0.112
Load sharing factor	[XGam]	1.000
Dynamic viscosity (mPa*s)	[etaM]	25.40 (65.0 °C)
Coefficient of friction	[mym]	0.073

Integral temperature-criteria

Tooth mass temperature (°C)	[theMC]	83.72
(theMC = theoil + XS*0.70*theflaint)		
Mean flash temperature (°C)	[theflaint]	22.29
Integral scuffing temperature (°C)	[theSint]	528.15
Flash factor (°K*N^-.75*s^.5*m^-.5*mm)	[XM]	50.058
Contact ratio factor	[Xeps]	0.270
Dynamic viscosity (mPa*s)	[etaOil]	88.12 (65.0 °C)
Mean coefficient of friction	[mym]	0.057
Geometry factor	[XBE]	0.218
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.291
Integral tooth flank temperature (°C)	[theint]	117.16

6. MEASUREMENTS FOR TOOTH THICKNESS

		------- Gear 1	------ Gear 2 --
Tooth thickness deviation		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.095 / -0.145	-0.130 / -0.190
Number of teeth spanned	[k]	5.000	15.000
Base tangent length (no backlash) (mm)	[Wk]	69.752	223.455
Actual base tangent length ('span') (mm)	[Wk.e/i]	69.663 / 69.616	223.333 / 223.277
(mm)	[ΔWk.e/i]	-0.089 / -0.136	-0.122 / -0.179
Diameter of contact point (mm)	[dMWk.m]	143.058	551.445
> Base tangent length Gear 2 is not measurable (Gear to thin)			
Theoretical diameter of ball/pin (mm)	[DM]	8.895	8.405
Effective diameter of ball/pin (mm)	[DMeff]	9.000	9.000
Radial single-ball measurement backlash free (mm)	[MrK]	76.856	282.880

Radial single-ball measurement (mm)	[MrK.e/i]	76.752 / 76.697	282.710 / 282.632
Diameter of contact point (mm)	[dMMr.m]	140.810	553.161
Diametral measurement over two balls without clearance (mm)	[MdK]	153.427	565.760
Diametral two ball measure (mm)	[MdK.e/i]	153.219 / 153.108	565.420 / 565.263
Diametral measurement over pins without clearance (mm)	[MdR]	153.713	565.760
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	153.504 / 153.393	565.420 / 565.263
Measurement over 2 pins (free) according to AGMA 2002 (mm)	[dk2f.e/i]	153.130 / 153.020	0.000 / 0.000
Measurement over 2 pins (transverse) according to AGMA 2002 (mm)	[dk2t.e/i]	153.782 / 153.672	0.000 / 0.000
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	153.504 / 153.393	565.420 / 565.263
Chordal tooth thickness (no backlash) (mm)	[sc]	8.929	8.027
Actual chordal tooth thickness (mm)	[sc.e/i]	8.834 / 8.784	7.897 / 7.837
Reference chordal height from da.m (mm)	[ha]	6.573	5.234
Tooth thickness (Arc) (mm)	[sn]	8.933	8.027
(mm)	[sn.e/i]	8.838 / 8.788	7.897 / 7.837
Backlash free center distance (mm)	[aControl.e/i]	346.199 /346.051	
Backlash free center distance, allowances (mm)	[jta]	-0.301 / -0.449	
dNf.i with aControl (mm)	[dNf0.i]	131.802	543.939
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	-0.040	0.786
Tip clearance (mm)	[c0.i(aControl)]	0.980	0.932
Centre distance allowances (mm)	[Aa.e/i]	0.029 / -0.029	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.024 / -0.024	
Radial clearance (mm)	[jrw]	0.477 / 0.272	
Circumferential backlash (transverse section) (mm)	[jtw]	0.395 / 0.226	
Normal backlash (mm)	[jnw]	0.337 / 0.192	
Torsional angle at entry with fixed output:			
Entire torsional angle (°)	[j.tSys]	0.0817/0.0467	

7. GEAR ACCURACY

		------- GEAR 1	-------- GEAR 2 --
According to DIN 3961:1978			
Accuracy grade	[Q-DIN3961]	6	6
Profile form deviation (μm)	[ff]	10.00	10.00
Profile slope deviation (μm)	[fHa]	7.00	7.00
Total profile deviation (μm)	[Ff]	13.00	13.00
Helix form deviation (μm)	[fbf]	8.00	8.00
Helix slope deviation (μm)	[fHb]	10.00	10.00
Total helix deviation (μm)	[Fb]	13.00	13.00
Normal base pitch deviation (μm)	[fpe]	9.00	10.00
Single pitch deviation (μm)	[fp]	9.00	10.00
Adjacent pitch difference (μm)	[fu]	11.00	12.00
Total cumulative pitch deviation (μm)	[Fp]	35.00	40.00
Sector pitch deviation over z/8 pitches (μm)	[Fpz/8]	22.00	25.00
Runout (μm)	[Fr]	25.00	28.00
Tooth Thickness Variation (μm)	[Rs]	15.00	16.00
Single flank composite, total (μm)	[Fi']	38.00	42.00
Single flank composite, tooth-to-tooth (μm)	[fi']	15.00	16.00
Radial composite, total (μm)	[Fi"]	28.00	32.00
Radial composite, tooth-to-tooth (μm)	[fi"]	12.00	14.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality)
6)

Maximum value for deviation error of axis (µm)	[fSigbet]	29.88 (Fb=15.00)
Maximum value for inclination error of axes (µm)	[fSigdel]	59.76

8. ADDITIONAL DATA

Mass (kg)	[m]	15.223	237.020
Total mass (kg)	[m]	252.243	

Moment of inertia (system with reference to the drive):
calculation without consideration of the exact tooth shape

single gears ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.03709	8.99133
System ((da+df)/2...di) (kg*m²)	[TraeghMom]	9.58475	

Torsional stiffness at entry with driven force fixed:

Torsional stiffness (MNm/rad)	[cr]	121.181
Torsion when subjected to nominal torque (°)	[delcr]	0.012
Mean coeff. of friction (acc. Niemann)	[mum]	0.039
Wear sliding coef. by Niemann	[zetw]	0.596
Gear power loss (kW)	[PVZ]	4.112
(Meshing efficiency (%)	[etaz]	99.589)
Sound pressure level (according to Masuda)	[dB(A)]	100.3
Oil requirement for injection lubrication (l/min)	[Voil]	12.952

(with oil cooler, for assumed difference in temperature of oil (°C):
10)

Classification according to F.E.M. (Edition 1.001, 1998)

Spectrum factor	[km]	0.090
Spectrum class	[L]	1
Application class (predefined service life)	[T]	9
Machine class (predefined service life)	[M]	8
Application class (achievable service life)	[T]	9
Machine class (achievable service life)	[M]	8

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Profile and tooth trace modifications for gear 1

Symmetric (both flanks)

- Tip relief, linear Caa = 50.000µm LCa = 0.718*mn dCa = 147.163mm

Profile and tooth trace modifications for gear 2

Symmetric (both flanks)

- Tip relief, linear Caa = 50.000µm LCa = 0.718*mn dCa = 559.188mm

Tip relief verification

Diameter (mm)	[dcheck]	150.726	562.005
Tip relief left/right (µm)	[Ca L/R]	48.6 / 48.6	48.2 / 48.2

Data for the tooth form calculation :
Data not available.

10. SERVICE LIFE, DAMAGE

Calculation with load spectrum

Required safety for tooth root	[SFmin]		1.56
Required safety for tooth flank	[SHmin]		1.25

Service life (calculated with required safeties):

System service life (h)	[Hatt]		205201
Tooth root service life (h)	[HFatt]	1e+006	2.052e+005
Tooth flank service life (h)	[HHatt]	1e+006	1e+006

Note: The entry 1e+006 h means that the Service life > 1,000,000 h.

Damage calculated on the basis of the required service life [H] (175200.0 h)

No.	F1%	F2%	H1%	H2%
1	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00
9	0.00	0.00	0.00	0.00
10	0.00	84.63	9.34	4.67
11	0.84	0.75	0.08	0.04
Σ	0.84	85.38	9.42	4.71

Damage calculated on basis of system service life [Hatt] (205200.7 h)

No.	F1%	F2%	H1%	H2%
1	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00
9	0.00	0.00	0.00	0.00
10	0.00	99.12	10.94	5.47
11	0.98	0.88	0.09	0.05
Σ	0.98	100.00	11.04	5.52

Damage calculated on basis of individual service life HFatt & HHatt

	HFatt1	HFatt2	HHatt1	HHatt2
(h)	1e+006	2.052e+005	1e+006	1e+006
No.	F1%	F2%	H1%	H2%
1	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00
9	0.00	0.00	0.00	0.00
10	0.00	99.12	99.15	99.15

11	100.00	0.88	0.85	0.85
Σ	100.00	100.00	100.00	100.00

Most critical duty cycle elements for Scoring (SB, Sint), Tooth Flank Fracture (SFF), hardened layer (SEHT) and Micropitting (Slam)

Calculation of the factors required to define reliability R(t) according to B. Bertsche with Weibull distribution:
R(t) = 100 * Exp(-((t*fac - t0)/(T - t0))^b) %; t (h)

Gear		fac	b	t0	T	R(H)%
1	Tooth root	89723	1.7	1.811e+012	2.783e+012	100.00
1	Tooth flank	89723	1.3	1.504e+011	7.164e+011	100.00
2	Tooth root	44862	1.7	8.887e+009	1.366e+010	100.00
2	Tooth flank	44862	1.3	1.504e+011	7.164e+011	100.00

Reliability of the configuration for required service life (%) 100.00 (Bertsche)

REMARKS:
- Specifications with [.e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances
  Specifications with [.m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances
  The calculation is done for the operating pitch circle.
- Details of calculation method:
  cg according to method B

End of Report lines: 638

A.2 KISSsoft Reports - Shaft calculation

A.2.1 KISSsoft Report - Planetary coaxial shafts (considering nominal load)

KISSsoft Release 03/2017 A

KISSsoft University license - Universidade do Porto

File

Name : Unnamed

Changed by: Joana Mêda de Sousa on: 10.10.2017 at: 00:03:14

Important hint: At least one warning has occurred during the calculation:

1-> Shaft 'PlanetCarrierShaft', Rolling bearing 'PlanetCarrierRolBearingRS':
The minimal load of the bearing is not achieved!
(P = 18.6 kN, Pmind = 36.6 kN, Condition: P/C > 2.000 %)

2-> Shaft 'SunShaft', Rolling bearing 'SunRolBearingGS2':
Axial cylindrical roller bearing:
This bearing can only accept axial forces in one direction.

3-> Cross section B-B:
Please note: The static maximum rate of utilization is greater than the dynamic rate,
so the static strength verification is obligatory!

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	PlanetCarrierShaft
Drawing	
Initial position (mm)	0.000
Length (mm)	1183.000
Speed (1/min)	15.00
Sense of rotation: clockwise	
Material	EN-GJS-400-15 (GGG 40)
Young's modulus (N/mm²)	169000.000
Poisson's ratio nu	0.275
Density (kg/m³)	7100.000
Coefficient of thermal expansion (10^-6/K)	12.500
Temperature (°C)	40.000
Weight of shaft (kg)	3320.518
Weight of shaft, including additional masses (kg)	3320.518
Mass moment of inertia (kg*m²)	1019.848
Momentum of mass GD2 (Nm²)	40018.823
Label	RingShaft
Drawing	
Initial position (mm)	687.500
Length (mm)	265.000
Speed (1/min)	0.00
Sense of rotation: clockwise	
Material	42 CrMo 4 (2)

Young's modulus (N/mm²)		206000.000
Poisson's ratio nu		0.300
Density (kg/m³)		7830.000
Coefficient of thermal expansion	(10^-6/K)	11.500
Temperature (°C)		40.000
Weight of shaft (kg)		1624.318
Weight of shaft, including additional masses (kg)		1624.318
Mass moment of inertia (kg*m²)		1607.247
Momentum of mass GD2 (Nm²)		63068.367

Label	SunShaft
Drawing	
Initial position (mm)	687.500
Length (mm)	1351.000
Speed (1/min)	77.88

Sense of rotation: clockwise

Material		18CrNiMo7-6
Young's modulus (N/mm²)		206000.000
Poisson's ratio nu		0.300
Density (kg/m³)		7830.000
Coefficient of thermal expansion	(10^-6/K)	11.500
Temperature (°C)		40.000
Weight of shaft (kg)		540.307
Weight of shaft, including additional masses (kg)		540.307
Mass moment of inertia (kg*m²)		9.199
Momentum of mass GD2 (Nm²)		360.954

Weight towards (0.000, 0.000, -1.000)

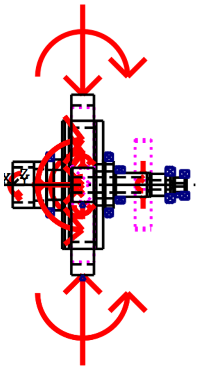
Consider deformations due to shearing

Shear correction coefficient 1.100

Rolling bearing stiffness is calculated from inner bearing geometry

Tolerance field: Mean value

Housing material		EN-GJS-400-15 (GGG 40)
Coefficient of thermal expansion	(10^-6/K)	12.500
Temperature of housing (°C)		40.000
Thermal housing reference point (mm)		0.000
Reference temperature (°C)		20.000



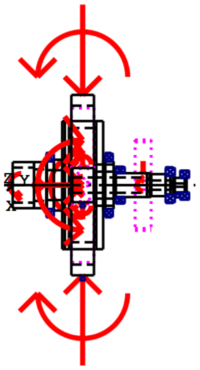


Figure: Load applications

Shaft definition (PlanetCarrierShaft)

Outer contour

Cylinder (Cylinder)			0.000mm ... 395.000mm
Diameter (mm)	[d]	540.0000	
Length (mm)	[l]	395.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder (Cylinder)			395.000mm ... 482.000mm
Diameter (mm)	[d]	558.8000	
Length (mm)	[l]	87.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder (Cylinder)			482.000mm ... 580.000mm
Diameter (mm)	[d]	595.0000	
Length (mm)	[l]	98.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder (Cylinder)			580.000mm ... 1060.000mm
Diameter (mm)	[d]	1500.0000	
Length (mm)	[l]	480.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder (Cylinder)			1060.000mm ... 1085.000mm
Diameter (mm)	[d]	595.0000	
Length (mm)	[l]	25.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder (Cylinder)			1085.000mm ... 1183.000mm
Diameter (mm)	[d]	558.8000	
Length (mm)	[l]	98.0000	
Surface roughness (µm)	[Rz]	8.0000	

Inner contour

Cylinder inside (Cylindrical bore)			0.000mm ... 250.000mm
Diameter (mm)	[d]	410.0000	
Length (mm)	[l]	250.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder inside (Cylinder inside)			250.000mm ... 400.000mm
Diameter (mm)	[d]	370.0000	
Length (mm)	[l]	150.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder inside (Cylinder inside)			400.000mm ... 610.000mm
Diameter (mm)	[d]	350.0000	
Length (mm)	[l]	210.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder inside (Cylinder inside)			610.000mm ... 662.000mm
Diameter (mm)	[d]	30.0000	
Length (mm)	[l]	52.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder inside (Cylinder inside)			662.000mm ... 978.000mm
Diameter (mm)	[d]	1337.0000	
Length (mm)	[l]	316.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder inside (Cylinder inside)			978.000mm ... 1183.000mm
Diameter (mm)	[d]	483.0000	
Length (mm)	[l]	205.0000	
Surface roughness (µm)	[Rz]	8.0000	

Forces

Type of force element		**Coupling**
Label in the model		CouplingPlanetCarrier(PlanetCarrierShaft)
Position on shaft (mm)	[y_{local}]	100.0000
Position in global system (mm)	[y_{global}]	100.0000
Effective diameter (mm)		540.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		200.0000
Power (kW)		2000.0007 driven (Input)
Torque (Nm)		1273240.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia Jp (kg*m²)		0.0000

Mass moment of inertia Jxx (kg*m²)		0.0000
Mass moment of inertia Jzz (kg*m²)		0.0000
Eccentricity (mm)		0.0000

Type of force element		**Coupling**
Label in the model		PlanetCarrierCoupling(PlanetCarrierConstraint)
Position on shaft (mm)	[y_{local}]	820.0000
Position in global system (mm)	[y_{global}]	820.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		256.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia Jp (kg*m²)		0.0000
Mass moment of inertia Jxx (kg*m²)		0.0000
Mass moment of inertia Jzz (kg*m²)		0.0000
Eccentricity (mm)		0.0000

Type of force element		**Coupling**
Label in the model		PlanetCarrierCoupling(PlanetRingGearingConstraint)
Position on shaft (mm)	[y_{local}]	820.0000
Position in global system (mm)	[y_{global}]	820.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		256.0000
Power (kW)		1007.4078 driving (Output)
Torque (Nm)		-641335.7037
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia Jp (kg*m²)		0.0000
Mass moment of inertia Jxx (kg*m²)		0.0000
Mass moment of inertia Jzz (kg*m²)		0.0000
Eccentricity (mm)		0.0000

Type of force element		**Coupling**
Label in the model		PlanetCarrierCoupling(SunPlanetGearingConstraint)
Position on shaft (mm)	[y_{local}]	820.0000
Position in global system (mm)	[y_{global}]	820.0000
Effective diameter (mm)		170.0000
Radial force factor (-)		0.0000
Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		256.0000

Power (kW)	992.5929 driving (Output)
Torque (Nm)	-631904.2963
Axial force (N)	0.0000
Shearing force X (N)	0.0000
Shearing force Z (N)	0.0000
Bending moment X (Nm)	0.0000
Bending moment Z (Nm)	0.0000
Mass (kg)	0.0000
Mass moment of inertia Jp (kg*m²)	0.0000
Mass moment of inertia Jxx (kg*m²)	0.0000
Mass moment of inertia Jzz (kg*m²)	0.0000
Eccentricity (mm)	0.0000

Type of force element		**Centric force**
Label in the model		Centric force
Position on shaft (mm)	$[y_{local}]$	360.0000
Position in global system (mm)	$[y_{global}]$	360.0000
Length of load application (mm)		0.0000
Power (kW)		0.0000
Torque (Nm)		-0.0000
Axial force (N)		20000.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

Bearing

Label in the model	PlanetCarrierRolBearingGS
Bearing type	SKF EE 843220/843290
Bearing type	Taper roller bearing (single row)

Bearing position (mm)	$[y_{lokal}]$	1129.054
Bearing position (mm)	$[y_{global}]$	1129.054
Attachment of external ring		Set fixed bearing right
Inner diameter (mm)	[d]	558.800
External diameter (mm)	[D]	736.600
Width (mm)	[b]	88.108
Corner radius (mm)	[r]	6.400
Number of rolling bodies	[Z]	42
Rolling body reference circle (mm)	$[D_{pw}]$	649.837
Diameter rolling body (mm)	$[D_w]$	42.741
Rolling body length (mm)	$[L_{we}]$	57.411
Distance a (mm)	[a]	111.000
Diameter, external race (mm)	$[d_o]$	691.756
Diameter, internal race (mm)	$[d_i]$	607.919

Calculation with approximate bearings internal geometry (*)

Bearing clearance 0.00 µm

The bearing pressure angle will be considered in the calculation

Position (center of pressure) (mm)
1062.1080

Basic static load rating (kN)	$[C_0]$	4150.000
Basic dynamic load rating (kN)	[C]	1830.000
Fatigue load rating (kN)	$[C_u]$	305.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	$[C_{theo}]$	1830.818
Basic static load rating (kN)	$[C_{0theo}]$	4150.189

Label in the model		PlanetCarrierRolBearingRS
Bearing type		SKF EE 843220/843290
Bearing type		Taper roller bearing (single row)

Bearing position (mm)	[y_{lokal}]	437.946
Bearing position (mm)	[y_{global}]	437.946
Attachment of external ring		Set fixed bearing left
Inner diameter (mm)	[d]	558.800
External diameter (mm)	[D]	736.600
Width (mm)	[b]	88.108
Corner radius (mm)	[r]	6.400
Number of rolling bodies	[Z]	42
Rolling body reference circle (mm)	[D_{pw}]	649.837
Diameter rolling body (mm)	[D_w]	42.741
Rolling body length (mm)	[L_{we}]	57.411
Distance a (mm)	[a]	111.000
Diameter, external race (mm)	[d_o]	691.756
Diameter, internal race (mm)	[d_i]	607.919

Calculation with approximate bearings internal geometry (*)

Bearing clearance 0.00 µm

The bearing pressure angle will be considered in the calculation

Position (center of pressure) (mm)

504.8920

Basic static load rating (kN)	[C_0]	4150.000
Basic dynamic load rating (kN)	[C]	1830.000
Fatigue load rating (kN)	[C_u]	305.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	1830.818
Basic static load rating (kN)	[C_{0theo}]	4150.189

Shaft definition (RingShaft)

Outer contour

Cylinder (Cylinder) 0.000mm ... 265.000mm

Diameter (mm)	[d]	2111.0000
Length (mm)	[l]	265.0000
Surface roughness (µm)	[Rz]	8.0000

Inner contour

Cylinder inside (Cylindrical bore) 0.000mm ... 265.000mm

Diameter (mm)	[d]	1860.0000
Length (mm)	[l]	265.0000
Surface roughness (µm)	[Rz]	8.0000

Forces

Type of force element		**Coupling**
Label in the model		CouplingRing(RingShaft)
Position on shaft (mm)	[y_{local}]	132.5000
Position in global system (mm)	[y_{global}]	820.0000
Effective diameter (mm)		2111.0000
Radial force factor (-)		0.0000

Direction of the radial force (°)		0.0000
Axial force factor (-)		0.0000
Length of load application (mm)		265.0000
Power (kW)		0.0000 driven (Input)
Torque (Nm)		1028023.4070
Axial force (N)		0.0000
Shearing force X (N)		0.0000
Shearing force Z (N)		0.0000
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000
Mass (kg)		0.0000
Mass moment of inertia Jp (kg*m²)		0.0000
Mass moment of inertia Jxx (kg*m²)		0.0000
Mass moment of inertia Jzz (kg*m²)		0.0000
Eccentricity (mm)		0.0000

Type of force element		**Cylindrical gear**
Label in the model		RingGear(PlanetRingGearingConstraint)
Position on shaft (mm)	[y_{local}]	132.5000
Position in global system (mm)	[y_{global}]	820.0000
Operating pitch diameter (mm)		-1814.5295
Helix angle (°)		15.0714 right
Working pressure angle at normal section (°)		20.7167
Position of contact (°)		0.0000
Length of load application (mm)		265.0000
Power (kW)		0.0000 driving (Output)
Torque (Nm)		-342674.4691
Axial force (N)		101709.4941
Shearing force X (N)		147935.6464
Shearing force Z (N)		377700.6403
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		92277.4366

Type of force element		**Cylindrical gear**
Label in the model		RingGear(PlanetRingGearingConstraint)2
Position on shaft (mm)	[y_{local}]	132.5000
Position in global system (mm)	[y_{global}]	820.0000
Operating pitch diameter (mm)		-1814.5295
Helix angle (°)		15.0714 right
Working pressure angle at normal section (°)		20.7167
Position of contact (°)		120.0000
Length of load application (mm)		265.0000
Power (kW)		0.0000 driving (Output)
Torque (Nm)		-342674.4691
Axial force (N)		101709.4941
Shearing force X (N)		-401066.1727
Shearing force Z (N)		-60734.2923
Bending moment X (Nm)		-79914.6043
Bending moment Z (Nm)		-46138.7183

Type of force element		**Cylindrical gear**
Label in the model		RingGear(PlanetRingGearingConstraint)3
Position on shaft (mm)	[y_{local}]	132.5000
Position in global system (mm)	[y_{global}]	820.0000
Operating pitch diameter (mm)		-1814.5295
Helix angle (°)		15.0714 right
Working pressure angle at normal section (°)		20.7167

Position of contact (°)	240.0000
Length of load application (mm)	265.0000
Power (kW)	0.0000 driving (Output)
Torque (Nm)	-342674.4691
Axial force (N)	101709.4941
Shearing force X (N)	253130.5264
Shearing force Z (N)	-316966.3480
Bending moment X (Nm)	79914.6043
Bending moment Z (Nm)	-46138.7183

Bearing

Label in the model	Support
Bearing type	Own Input

Bearing position (mm)	[y_{lokal}]	132.500
Bearing position (mm)	[y_{global}]	820.000

Degrees of freedom

X: fixedY: fixedZ: fixed

Rx: fixedRy: fixedRz: fixed

Shaft definition (SunShaft)

Outer contour

Cylinder (Cylinder) 0.000mm ... 265.000mm

Diameter (mm)	[d]	400.0000
Length (mm)	[l]	265.0000
Surface roughness (µm)	[Rz]	8.0000

Cylinder (Cylinder) 265.000mm ... 520.000mm

Diameter (mm)	[d]	315.0000
Length (mm)	[l]	255.0000
Surface roughness (µm)	[Rz]	8.0000

Radius left (Radius left)

r=15.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder) 520.000mm ... 935.000mm

Diameter (mm)	[d]	280.0000
Length (mm)	[l]	415.0000
Surface roughness (µm)	[Rz]	8.0000

Square groove (Square groove)

b=5.00 (mm), t=4.00 (mm), r=0.50 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Relief groove left (Relief groove left)

r=1.20 (mm), t=0.40 (mm), l=4.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Form F (DIN 509), Series 1, with the usual stressing

Cylinder (Cylinder) 935.000mm ... 1105.000mm

Diameter (mm)	[d]	257.0000
Length (mm)	[l]	170.0000
Surface roughness (µm)	[Rz]	8.0000

Radius left (Radius left)
r=1.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder)			1105.000mm ... 1211.000mm
Diameter (mm)	[d]	220.0000	
Length (mm)	[l]	106.0000	
Surface roughness (µm)	[Rz]	8.0000	

Relief groove left (Relief groove left)
r=1.20 (mm), t=0.40 (mm), l=4.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)
Form F (DIN 509), Series 1, with the usual stressing

Cylinder (Cylinder)			1211.000mm ... 1351.000mm
Diameter (mm)	[d]	170.0000	
Length (mm)	[l]	140.0000	
Surface roughness (µm)	[Rz]	8.0000	

Radius left (Radius left)
r=5.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Inner contour

Cone inside (Cone inside)			0.000mm ... 10.000mm
Diameter left (mm)	[d_l]	200.0000	
Diameter right (mm)	[d_r]	180.0000	
Length (mm)	[l]	10.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder inside (Cylinder inside)			10.000mm ... 885.000mm
Diameter (mm)	[d]	180.0000	
Length (mm)	[l]	875.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cone inside (Cone inside)			885.000mm ... 920.000mm
Diameter left (mm)	[d_l]	180.0000	
Diameter right (mm)	[d_r]	110.0000	
Length (mm)	[l]	35.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder inside (Cylinder inside)			920.000mm ... 1160.000mm
Diameter (mm)	[d]	110.0000	
Length (mm)	[l]	240.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cone inside (Cone inside)			1160.000mm ... 1195.500mm
Diameter left (mm)	[d_l]	110.0000	
Diameter right (mm)	[d_r]	39.0000	
Length (mm)	[l]	35.5000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder inside (Cylinder inside)			1195.500mm ... 1331.000mm
Diameter (mm)	[d]	39.0000	
Length (mm)	[l]	135.5000	
Surface roughness (µm)	[Rz]	8.0000	

Forces

Type of force element		**Cylindrical gear**
Label in the model		LowSpeedGear(InfLowSpeedIntermediateGearingConstraint)
Position on shaft (mm)	[y_{local}]	841.0000
Position in global system (mm)	[y_{global}]	1528.5000
Operating pitch diameter (mm)		1063.4483
Helix angle (°)		25.0871 Double helical gearing, left-right
Working pressure angle at normal section (°)		20.5050
Position of contact (°)		-150.0000
Length of load application (mm)		188.0000
Power (kW)		1000.0004 driving (Output)
Torque (Nm)		-122608.2963
Axial force (N)		0.0000
Shearing force X (N)		197754.6088
Shearing force Z (N)		-152084.4341
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

Type of force element		**Cylindrical gear**
Label in the model		LowSpeedGear(SupLowSpeedIntermediateGearingConstraint)
Position on shaft (mm)	[y_{local}]	841.0000
Position in global system (mm)	[y_{global}]	1528.5000
Operating pitch diameter (mm)		1063.4483
Helix angle (°)		25.0871 Double helical gearing, left-right
Working pressure angle at normal section (°)		20.5050
Position of contact (°)		150.0000
Length of load application (mm)		188.0000
Power (kW)		1000.0004 driving (Output)
Torque (Nm)		-122608.2963
Axial force (N)		0.0000
Shearing force X (N)		-32831.6791
Shearing force Z (N)		-247302.7320
Bending moment X (Nm)		0.0000
Bending moment Z (Nm)		0.0000

Type of force element		**Cylindrical gear**
Label in the model		SunGear(SunPlanetGearingConstraint)
Position on shaft (mm)	[y_{local}]	132.5000
Position in global system (mm)	[y_{global}]	820.0000
Operating pitch diameter (mm)		439.2836
Helix angle (°)		15.2859 left
Working pressure angle at normal section (°)		22.7023
Position of contact (°)		0.0000
Length of load application (mm)		265.0000
Power (kW)		666.6669 driven (Input)
Torque (Nm)		81738.8642
Axial force (N)		101709.4941
Shearing force X (N)		-161399.8748
Shearing force Z (N)		-372146.2206
Bending moment X (Nm)		-0.0000
Bending moment Z (Nm)		22339.6559

Type of force element		**Cylindrical gear**
Label in the model		SunGear(SunPlanetGearingConstraint)2
Position on shaft (mm)	[y_{local}]	132.5000
Position in global system (mm)	[y_{global}]	820.0000

Operating pitch diameter (mm)		439.2836
Helix angle (°)		15.2859 left
Working pressure angle at normal section (°)		22.7023
Position of contact (°)		120.0000
Length of load application (mm)		265.0000
Power (kW)		666.6669 driven (Input)
Torque (Nm)		81738.8642
Axial force (N)		101709.4941
Shearing force X (N)		402988.0184
Shearing force Z (N)		46296.7186
Bending moment X (Nm)		-19346.7096
Bending moment Z (Nm)		-11169.8280

Type of force element		**Cylindrical gear**
Label in the model		SunGear(SunPlanetGearingConstraint)3
Position on shaft (mm)	[y_{local}]	132.5000
Position in global system (mm)	[y_{global}]	820.0000
Operating pitch diameter (mm)		439.2836
Helix angle (°)		15.2859 left
Working pressure angle at normal section (°)		22.7023
Position of contact (°)		240.0000
Length of load application (mm)		265.0000
Power (kW)		666.6669 driven (Input)
Torque (Nm)		81738.8642
Axial force (N)		101709.4941
Shearing force X (N)		-241588.1436
Shearing force Z (N)		325849.5020
Bending moment X (Nm)		19346.7096
Bending moment Z (Nm)		-11169.8280

Bearing

Label in the model		SunRolBearingGS1
Bearing type		SKF NCF 2244 ECJB/PEX
Bearing type		Cylindrical roller bearing (single row)
		SKF Explorer
Bearing position (mm)	[y_{lokal}]	1159.000
Bearing position (mm)	[y_{global}]	1846.500
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	220.000
External diameter (mm)	[D]	400.000
Width (mm)	[b]	108.000
Corner radius (mm)	[r]	4.000
Number of rolling bodies	[Z]	12
Rolling body reference circle (mm)	[D_{pw}]	299.646
Diameter rolling body (mm)	[D_w]	69.438
Rolling body length (mm)	[L_{we}]	92.283
Diameter, external race (mm)	[d_o]	369.152
Diameter, internal race (mm)	[d_i]	230.140
Calculation with approximate bearings internal geometry (*)		
Bearing clearance		DIN 620:1988 C0 (135.00 µm)
Basic static load rating (kN)	[C_0]	2600.000
Basic dynamic load rating (kN)	[C]	2000.000
Fatigue load rating (kN)	[C_u]	240.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	1999.241
Basic static load rating (kN)	[C_{0theo}]	2599.348

Label in the model		SunRolBearingGS2
Bearing type		SKF 89434 M
Bearing type		Axial cylindrical roller bearing
Bearing position (mm)	[y_{lokal}]	1328.000
Bearing position (mm)	[y_{global}]	2015.500
Attachment of external ring		Axial bearing
Inner diameter (mm)	[d]	170.000
External diameter (mm)	[D]	340.000
Width (mm)	[b]	103.000
Corner radius (mm)	[r]	5.000
Number of rolling bodies	[Z]	12
Rolling body reference circle (mm)	[D_{pw}]	258.111
Diameter rolling body (mm)	[D_w]	33.534
Rolling body length (mm)	[L_{we}]	81.329
Diameter, external race (mm)	[d_o]	339.440
Diameter, internal race (mm)	[d_i]	176.782
Calculation with approximate bearings internal geometry (*)		
Bearing clearance		0.00 µm
Basic static load rating (kN)	[C_0]	7200.000
Basic dynamic load rating (kN)	[C]	1600.000
Fatigue load rating (kN)	[C_u]	600.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	1598.920
Basic static load rating (kN)	[C_{0theo}]	7200.000

Label in the model		SunRolBearingRS
Bearing type		SKF 23056 CCK/W33
Bearing type		Spherical roller bearings SKF Explorer
Bearing position (mm)	[y_{lokal}]	573.000
Bearing position (mm)	[y_{global}]	1260.500
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	280.000
External diameter (mm)	[D]	420.000
Width (mm)	[b]	106.000
Corner radius (mm)	[r]	4.000
Number of rolling bodies	[Z]	15
Rolling body reference circle (mm)	[D_{pw}]	360.360
Diameter rolling body (mm)	[D_w]	47.568
Rolling body length (mm)	[L_{we}]	52.825
Diameter, external race (mm)	[d_o]	407.580
Diameter, internal race (mm)	[d_i]	313.355
Radius of curvature, external race (mm)	[r_o]	206.229
Radius of curvature, internal race (mm)	[r_i]	206.229
Calculation with approximate bearings internal geometry (*)		
Bearing clearance		DIN 620:1988 C0 (215.00 µm)
Basic static load rating (kN)	[C_0]	2850.000
Basic dynamic load rating (kN)	[C]	1797.000
Fatigue load rating (kN)	[C_u]	224.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	1797.127
Basic static load rating (kN)	[C_{0theo}]	2850.117

Label in the model		SupportGS
Bearing type		Own Input

Bearing position (mm)	$[y_{lokal}]$	1245.000
Bearing position (mm)	$[y_{global}]$	1932.500

Degrees of freedom
X: freeY: freeZ: free
Rx: freeRy: freeRz: free

Label in the model		SupportRS
Bearing type		Own Input

Bearing position (mm)	$[y_{lokal}]$	132.500
Bearing position (mm)	$[y_{global}]$	820.000

Degrees of freedom
X: fixedY: freeZ: fixed
Rx: freeRy: freeRz: free

Results

Shaft

Maximum deflection 231.185 (µm) (SunShaft pos = 1538.000 mm)

Mass center of gravity	
PlanetCarrierShaft (mm)	744.056
RingShaft (mm)	132.500
SunShaft (mm)	511.084

Total axial load	
PlanetCarrierShaft (N)	20000.000
RingShaft (N)	305128.482
SunShaft (N)	305128.482

Torsion under torque	
PlanetCarrierShaft (°)	-0.077
RingShaft (°)	0.000
SunShaft (°)	-0.176

Bearing

Probability of failure	[n]	10.00	%
Axial clearance	$[u_A]$	10.00	µm
Lubricant	Oil: Castrol Optigear Synthetic X 320		

Lubricant with additive, effect on bearing lifetime confirmed in tests.
Oil lubrication, on-line filtration, ISO4406 -/17/14

Lubricant - service temperature	$[T_B]$	65.00	°C
Limit for factor aISO	$[a_{ISOmax}]$	50.00	
Oil level	$[h_{oil}]$	0.00	mm

Oil injection lubrication

Rolling bearing service life according to ISO/TS 16281:2008

Shaft 'PlanetCarrierShaft' Rolling bearing 'PlanetCarrierRolBearingGS'

Position (Y-coordinate)	[y]	1129.05	mm
Equivalent load	[P]	46.90	kN
Equivalent load	[P_0]	28.86	kN
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	13.952	
Nominal bearing service life	[L_{nh}]	> 1000000	h
Modified bearing service life	[L_{nmh}]	> 1000000	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	143.78	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	0.000	µm
Reference rating service life	[L_{nrh}]	> 1000000	h
Modified reference rating service life	[L_{nrmh}]	> 1000000	h
Effective static safety factor	[S_{0w}]	50.29	
Static safety factor	[S_{0ref}]	115.22	
Equivalent load	[P_{0ref}]	36.02	kN
Bearing reaction force	[Fx]	0.000	kN
Bearing reaction force	[Fy]	-24.294	kN
Bearing reaction force	[Fz]	14.000	kN
Bearing reaction force	[Fr]	14.000	kN (90°)
Bearing reaction moment	[Mx]	-988.67	Nm
Bearing reaction moment	[My]	0.00	Nm
Bearing reaction moment	[Mz]	0.00	Nm
Bearing reaction moment	[Mr]	988.67	Nm (180°)
Oil level	[H]	0.000	mm
Load-independent moment of friction	[M_0]	13.043	Nm
Load-dependent moment of friction	[M_1]	2.720	Nm
Moment of friction, cylindrical roller bearing[M_2]		0.000	Nm
Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994			
Torque of friction	[M_{loss}]	15.763	Nm
Power loss	[P_{loss}]	24.760	W
The moment of friction is calculated according to the details in SKF Catalog 1994.			
The factors used to calculate the torque loss have been assumed for this bearing.			
Displacement of bearing	[u_x]	0.000	µm
Displacement of bearing	[u_y]	334.150	µm
Displacement of bearing	[u_z]	0.791	µm
Displacement of bearing	[u_r]	0.791	µm (90°)
Misalignment of bearing	[r_x]	0.036	mrad (0.12')
Misalignment of bearing	[r_y]	-1.346	mrad (-4.63')
Misalignment of bearing	[r_z]	-0.000	mrad (0')
Misalignment of bearing	[r_r]	0.036	mrad (0.12')

Shaft 'PlanetCarrierShaft' Rolling bearing 'PlanetCarrierRolBearingRS'

Position (Y-coordinate)	[y]	437.95	mm
Equivalent load	[P]	18.57	kN
Equivalent load	[P_0]	18.57	kN
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	50.000	
Nominal bearing service life	[L_{nh}]	> 1000000	h
Modified bearing service life	[L_{nmh}]	> 1000000	h

Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	223.49	

Calculation with approximate bearings internal geometry

Operating bearing clearance	[Pd]	0.000	µm
Reference rating service life	[L_{nrh}]	> 1000000	h
Modified reference rating service life	[L_{nrmh}]	> 1000000	h
Effective static safety factor	[S_{0w}]	61.24	
Static safety factor	[S_{0ref}]	152.57	
Equivalent load	[P_{0ref}]	27.20	kN
Bearing reaction force	[Fx]	-0.000	kN
Bearing reaction force	[Fy]	4.294	kN
Bearing reaction force	[Fz]	18.569	kN
Bearing reaction force	[Fr]	18.569	kN (90°)
Bearing reaction moment	[Mx]	1282.98	Nm
Bearing reaction moment	[My]	0.00	Nm
Bearing reaction moment	[Mz]	0.00	Nm
Bearing reaction moment	[Mr]	1282.98	Nm (0°)
Oil level	[H]	0.000	mm
Load-independent moment of friction	[M_0]	13.043	Nm
Load-dependent moment of friction	[M_1]	3.608	Nm
Moment of friction, cylindrical roller bearing	[M_2]	0.000	Nm

Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994

Torque of friction	[M_{loss}]	16.651	Nm
Power loss	[P_{loss}]	26.155	W

The moment of friction is calculated according to the details in SKF Catalog 1994.

The factors used to calculate the torque loss have been assumed for this bearing.

Displacement of bearing	[u_x]	-0.000	µm
Displacement of bearing	[u_y]	161.772	µm
Displacement of bearing	[u_z]	-24.373	µm
Displacement of bearing	[u_r]	24.373	µm (-90°)
Misalignment of bearing	[r_x]	0.037	mrad (0.13')
Misalignment of bearing	[r_y]	-1.047	mrad (-3.6')
Misalignment of bearing	[r_z]	-0.000	mrad (0')
Misalignment of bearing	[r_r]	0.037	mrad (0.13')

Shaft 'RingShaft' Bearing 'Support'

Position (Y-coordinate)	[y]	132.50	mm
Bearing reaction force	[Fx]	0.000	kN
Bearing reaction force	[Fy]	-305.128	kN
Bearing reaction force	[Fz]	15.932	kN
Bearing reaction force	[Fr]	15.932	kN (90°)
Bearing reaction moment	[Mx]	0.00	Nm
Bearing reaction moment	[My]	0.00	Nm
Bearing reaction moment	[Mz]	0.00	Nm
Bearing reaction moment	[Mr]	0.00	Nm (44.97°)
Displacement of bearing	[u_x]	0.000	µm
Displacement of bearing	[u_y]	205.000	µm
Displacement of bearing	[u_z]	0.000	µm
Displacement of bearing	[u_r]	0.000	µm
Misalignment of bearing	[r_x]	0.000	mrad (0')
Misalignment of bearing	[r_y]	0.000	mrad (0')
Misalignment of bearing	[r_z]	0.000	mrad (0')
Misalignment of bearing	[r_r]	0.000	mrad (0')

Shaft 'SunShaft' Rolling bearing 'SunRolBearingGS1'

Position (Y-coordinate)	[y]	1159.00	mm
Equivalent load	[P]	263.26	kN
Equivalent load	[P_0]	263.26	kN
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	3.688	
Nominal bearing service life	[L_{nh}]	184458.69	h
Modified bearing service life	[L_{nmh}]	680334.97	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	9.88	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	135.000	µm
Reference rating service life	[L_{nrh}]	375726.92	h
Modified reference rating service life	[L_{nrmh}]	813451.16	h
Effective static safety factor	[S_{0w}]	6.82	
Static safety factor	[S_{0ref}]	8.01	
Equivalent load	[P_{0ref}]	324.42	kN
Bearing reaction force	[Fx]	-99.867	kN
Bearing reaction force	[Fy]	0.000	kN
Bearing reaction force	[Fz]	243.579	kN
Bearing reaction force	[Fr]	263.257	kN (112.29°)
Bearing reaction moment	[Mx]	-353.60	Nm
Bearing reaction moment	[My]	0.00	Nm
Bearing reaction moment	[Mz]	-143.30	Nm
Bearing reaction moment	[Mr]	381.54	Nm (-157.94°)
Oil level	[H]	0.000	mm
Load-independent moment of friction	[M_0]	3.685	Nm
Load-dependent moment of friction	[M_1]	32.644	Nm
Moment of friction, cylindrical roller bearing[M_2]		0.000	Nm
Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994			
Torque of friction	[M_{loss}]	36.329	Nm
Power loss	[P_{loss}]	296.301	W
The moment of friction is calculated according to the details in SKF Catalog 1994.			
Displacement of bearing	[u_x]	65.978	µm
Displacement of bearing	[u_y]	507.744	µm
Displacement of bearing	[u_z]	-155.787	µm
Displacement of bearing	[u_r]	169.183	µm (-67.05°)
Misalignment of bearing	[r_x]	0.169	mrad (0.58')
Misalignment of bearing	[r_y]	-3.066	mrad (-10.54')
Misalignment of bearing	[r_z]	0.068	mrad (0.23')
Misalignment of bearing	[r_r]	0.182	mrad (0.63')

Shaft 'SunShaft' Rolling bearing 'SunRolBearingGS2'

Position (Y-coordinate)	[y]	1328.00	mm
Equivalent load	[P]	305.13	kN
Equivalent load	[P_0]	305.13	kN
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	1.969	
Nominal bearing service life	[L_{nh}]	53602.85	h
Modified bearing service life	[L_{nmh}]	105537.77	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	23.60	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	0.000	µm

Reference rating service life	[L_{nrh}]	157148.95	h
Modified reference rating service life	[L_{nrmh}]	183103.02	h
Effective static safety factor	[S_{0w}]	11.61	
Static safety factor	[S_{0ref}]	16.04	
Equivalent load	[P_{0ref}]	448.90	kN
Bearing reaction force	[Fx]	0.000	kN
Bearing reaction force	[Fy]	-305.128	kN
Bearing reaction force	[Fz]	0.000	kN
Bearing reaction force	[Fr]	0.000	kN
Bearing reaction moment	[Mx]	-6783.84	Nm
Bearing reaction moment	[My]	0.00	Nm
Bearing reaction moment	[Mz]	-2739.12	Nm
Bearing reaction moment	[Mr]	7315.96	Nm (-158.01°)
Oil level	[H]	0.000	mm
Load-independent moment of friction	[M_0]	2.051	Nm
Load-dependent moment of friction	[M_1]	116.712	Nm
Moment of friction, cylindrical roller bearing[M_2]		0.000	Nm

Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994

Torque of friction	[M_{loss}]	118.763	Nm
Power loss	[P_{loss}]	968.636	W

The moment of friction is calculated according to the details in SKF Catalog 1994.

Displacement of bearing	[u_x]	57.588	µm
Displacement of bearing	[u_y]	536.269	µm
Displacement of bearing	[u_z]	-135.013	µm
Displacement of bearing	[u_r]	146.782	µm (-66.9°)
Misalignment of bearing	[r_x]	0.059	mrad (0.2')
Misalignment of bearing	[r_y]	-3.066	mrad (-10.54')
Misalignment of bearing	[r_z]	0.024	mrad (0.08')
Misalignment of bearing	[r_r]	0.064	mrad (0.22')

Shaft 'SunShaft' Rolling bearing 'SunRolBearingRS'

Position (Y-coordinate)	[y]	573.00	mm
Equivalent load	[P]	103.20	kN
Equivalent load	[P_0]	103.20	kN
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	44.334	
Nominal bearing service life	[L_{nh}]	> 1000000	h
Modified bearing service life	[L_{nmh}]	> 1000000	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	27.62	

Calculation with approximate bearings internal geometry

Operating bearing clearance	[Pd]	215.000	µm
Reference rating service life	[L_{nrh}]	> 1000000	h
Modified reference rating service life	[L_{nrmh}]	> 1000000	h
Effective static safety factor	[S_{0w}]	9.51	
Static safety factor	[S_{0ref}]	19.15	
Equivalent load	[P_{0ref}]	148.86	kN
Bearing reaction force	[Fx]	-39.085	kN
Bearing reaction force	[Fy]	0.000	kN
Bearing reaction force	[Fz]	95.511	kN
Bearing reaction force	[Fr]	103.199	kN (112.26°)
Oil level	[H]	0.000	mm
Load-independent moment of friction	[M_0]	7.955	Nm
Load-dependent moment of friction	[M_1]	2.702	Nm
Moment of friction, cylindrical roller bearing[M_2]		0.000	Nm

Moment of friction for seals determined according to SKF main catalog 4000/IV T DE:1994

Torque of friction	[M_{loss}]	10.658	Nm
Power loss	[P_{loss}]	86.925	W

The moment of friction is calculated according to the details in SKF Catalog 1994.

Displacement of bearing	[u_x]	64.466	µm
Displacement of bearing	[u_y]	396.106	µm
Displacement of bearing	[u_z]	-154.674	µm
Displacement of bearing	[u_r]	167.571	µm (-67.37°)
Misalignment of bearing	[r_x]	-0.280	mrad (-0.96')
Misalignment of bearing	[r_y]	-1.412	mrad (-4.85')
Misalignment of bearing	[r_z]	-0.117	mrad (-0.4')
Misalignment of bearing	[r_r]	0.304	mrad (1.04')

Shaft 'SunShaft' Bearing 'SupportGS'

Position (Y-coordinate)	[y]	1245.00	mm
Bearing reaction force	[Fx]	0.000	kN
Bearing reaction force	[Fy]	0.000	kN
Bearing reaction force	[Fz]	0.000	kN
Bearing reaction force	[Fr]	0.000	kN
Displacement of bearing	[u_x]	60.686	µm
Displacement of bearing	[u_y]	522.896	µm
Displacement of bearing	[u_z]	-142.681	µm
Displacement of bearing	[u_r]	155.051	µm (-66.96°)
Misalignment of bearing	[r_x]	0.126	mrad (0.43')
Misalignment of bearing	[r_y]	-3.066	mrad (-10.54')
Misalignment of bearing	[r_z]	0.051	mrad (0.17')
Misalignment of bearing	[r_r]	0.136	mrad (0.47')

Shaft 'SunShaft' Bearing 'SupportRS'

Position (Y-coordinate)	[y]	132.50	mm
Bearing reaction force	[Fx]	-25.971	kN
Bearing reaction force	[Fy]	0.000	kN
Bearing reaction force	[Fz]	65.594	kN
Bearing reaction force	[Fr]	70.548	kN (111.6°)
Displacement of bearing	[u_x]	0.000	µm
Displacement of bearing	[u_y]	305.629	µm
Displacement of bearing	[u_z]	0.000	µm
Displacement of bearing	[u_r]	0.000	µm
Misalignment of bearing	[r_x]	-0.356	mrad (-1.23')
Misalignment of bearing	[r_y]	-0.043	mrad (-0.15')
Misalignment of bearing	[r_z]	-0.149	mrad (-0.51')
Misalignment of bearing	[r_r]	0.386	mrad (1.33')

(*) Note about roller bearings with an approximated bearing geometry:
The internal geometry of these bearings has not been input in the database.
The geometry is back-calculated as specified in ISO 281, from C and C0 (details in the manufacturer's catalog).
For this reason, the geometry may be different from the actual geometry.
This can lead to differences in the service life calculation and, more importantly, the roller bearing stiffness.

Damage (%) [Lreq] (175200.000)

Bin no	B1	B2	B3	B4	B5
1	0.02	0.00	21.54	95.68	0.57
Σ	0.02	0.00	21.54	95.68	0.57

Utilization (%) [Lreq] (175200.000)

B1	B2	B3	B4	B5
48.17	48.17	63.09	98.69	48.17

Note: Utilization = (Lreq/Lh)^(1/k)
Ball bearing: k = 3, roller bearing: k = 10/3

B1: PlanetCarrierRolBearingGS
B2: PlanetCarrierRolBearingRS
B3: SunRolBearingGS1
B4: SunRolBearingGS2
B5: SunRolBearingRS

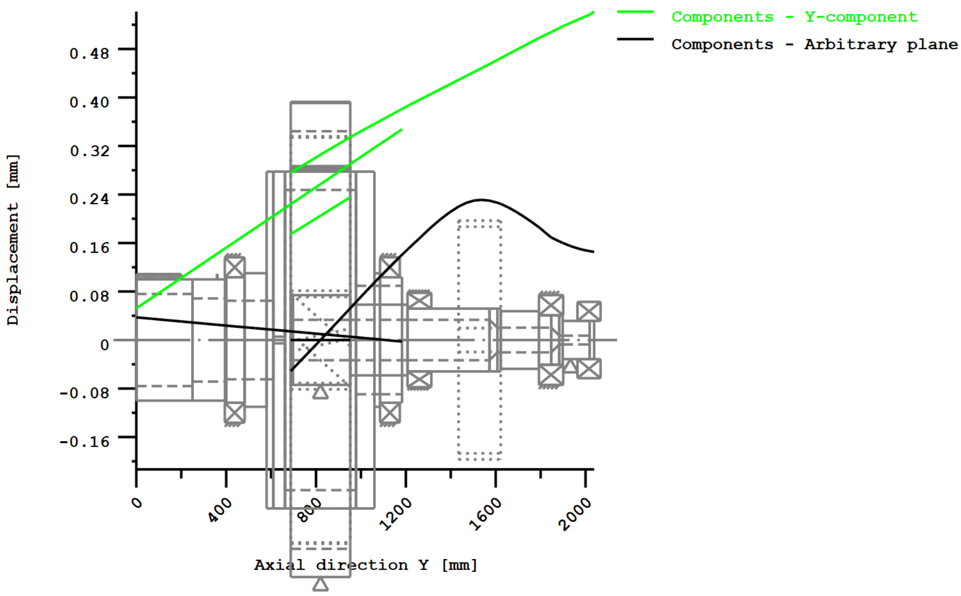
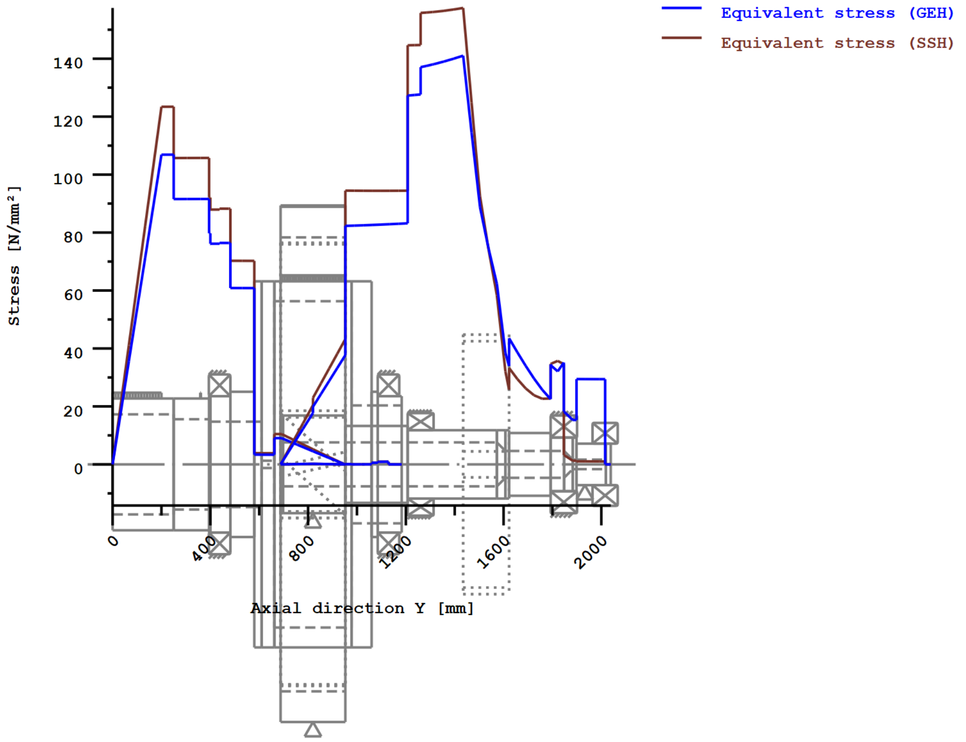


Figure: Deformation (bending etc.) (Arbitrary plane 292.6788741 121)



Nominal stresses, without taking into account stress concentrations
GEH(von Mises): sigV = ((sigB+sigZ,D)^2 + 3*(tauT+tauS)^2)^1/2
SSH(Tresca): sigV = ((sigB-sigZ,D)^2 + 4*(tauT+tauS)^2)^1/2

Figure: Equivalent stress

Strength calculation as specified in the FKM Guideline (6th Election, 2012)

Summary

SunShaft

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

Calculation of finite life fatigue strength and static strength

Rolled steel, case-hardening steel

Overload case F1 (chapter 2.4.2): Constant mean stress

Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 1.5	[jG]	1.00

Cross section	Pos (Y co-ord) (mm)	
A-A	935.00	Shoulder
B-B	628.50	Square groove
C-C	622.00	Interference fit

Results:

Cross section	Kfb	KRs	ALGmax	SD	SS	SB
A-A	3.43	0.89	0.42	3.19	8.40	11.86
B-B	3.43	1.00	0.32	4.24	2.39	3.37
C-C	2.63	1.00	0.21	6.37	2.67	3.77
Nominal safety:				1.35	1.40	1.85

Abbreviations:
Kfb: Notch factor bending
KRs: Surface factor
ALGmax: Highest utilization
SD: Safety endurance limit
SS: Safety against yield point
SB: Safety against tensile stress

Service life and damage

System service life (h)	[Hatt]	1000000.00
Damage to system (%)	[D]	0.00
Damage (%)	[H] (175200.0 h)	

Damage to shaft (%) [D]

SunShaft: 0.000

Calculation of reliability R(t) using a Weibull distribution; t in (h):

R(t) = 100 * Exp(-((t*fac - t0)/(T - t0))^b) %

Welle	fac	b	t0	T
3	900	1.5	8.207e+008	1.74e+009

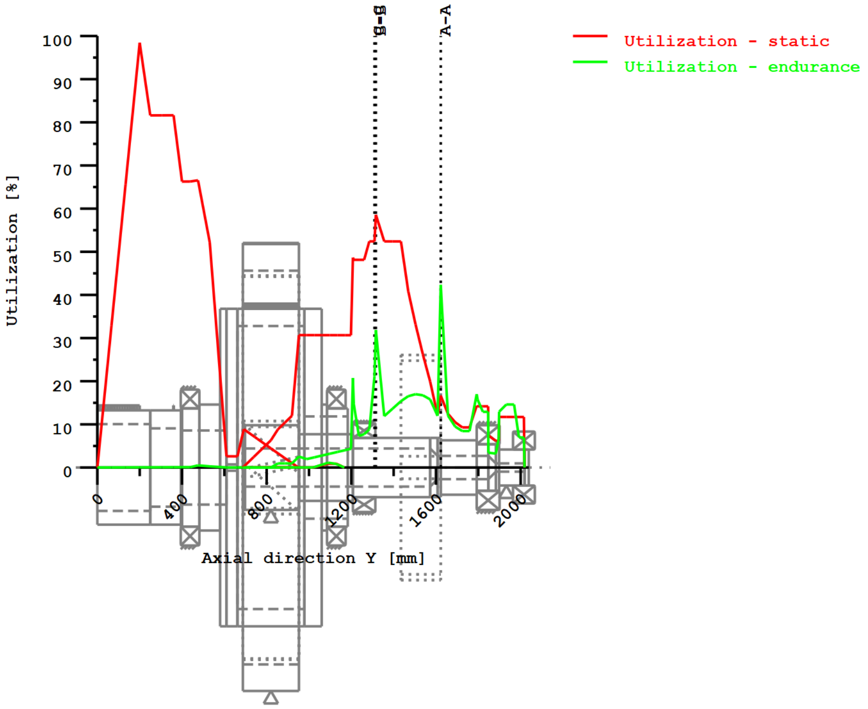
Damage to cross sections (%)	[D]
A-A:	0.00
B-B:	0.00
C-C:	0.00

Utilization (%) [Smin/S]

Cross section	Static	Endurance
A-A	16.662	42.332
B-B	58.588	31.864
C-C	52.409	21.184

Maximum utilization of shafts (%)

	[A]
SunShaft:	58.588



Utilization = Smin/S (%)

Figure: Strength

Calculation details

General statements

Label	SunShaft	
Drawing		
Length (mm)	[l]	1351.00
Speed (1/min)	[n]	77.88

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

	Tension/Compression	Bending	Torsion	Shearing
Load factor static calculation	1.700	1.700	1.700	1.700
Load factor endurance limit	1.000	1.000	1.000	1.000

Rolled steel, case-hardening steel

Base stress according FKM chapter 5.1:

Tensile strength (N/mm²)	[Rm,N]	1200.00	
Yield point (N/mm²)	[Rp,N]	850.00	
Fatigue limit (N/mm²)	[σzdWN]	480.00	
Fatigue limit (N/mm²)	[σbWN]	510.00	
Fatigue limit (N/mm²)	[τtWN]	305.00	
Fatigue limit (N/mm²)	[τsWN]	280.00	
Breaking elongation (%)	[A]	8.00	
Reference diameter (mm)	[deffNm, deffNp]	16.00	16.00

Required life time	[H]	175200.00
Number of load cycles	[NL]	818723077
Fatigue strength for single stage use		
Temperature (°C)	[Temperatur]	40.000
Temperature duration (h)	[TemperaturD]	175200.000

Temperature coefficients	[KTm, KTp, KTD]	1.000	1.000	1.000
	[KTtm, KTtp]	1.000	1.000	
Internal stress coefficient	[KEs, KEt]	1.000	1.000	
Additional coefficients	[KA, KW, KfW]	1.000	1.000	1.000
	[KNL, KNLE]	1.000	1.000	
Protective layer factor	[KS]	1.000		

Material properties:

[fσZ, fσD, fτ, Rpmax]	1.000	1.000	0.577	1150.0
[fWt, fWs]	0.577	0.400		
[aM, bM, aTD]	0.35000	-0.100	1.400	
[aG, bG, aRsig, RmNmin]	0.500	2700.0	0.220	400.0
[MS, MT]	0.1627	0.0939		
[kσ, kτ]	15	25		
[kDσ, kDτ]	0	0		
[NDσ, NDτ]	1e+006	1e+006		
[NDσII, NDτII]	0	0		

Thickness of raw material (mm)	[d.eff]	410.00	
Material data calculated acc. FKM directive with Kdm, Kdp			
Geometric size factors (Kdm, Kdp) calculated from raw diameter			
Material strength calculated from size of raw material			
Constants	[adm, adp]	0.370	0.370
Size factors	[Kdm, Kdp]	0.625	0.625
Tensile strength (N/mm²)	[Rm]	750.54	
Yield point (N/mm²)	[Rp]	531.63	
σzdW (N/mm²)	[σzdW]	300.22	
σbW (N/mm²)	[σbW]	318.98	
τtW (N/mm²)	[τtW]	190.76	
τsW (N/mm²)	[τsW]	175.13	

Overload case F1 (chapter 2.4.2): Constant mean stress

Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jG]	1.00

Cross section 'A-A' Shoulder

Comment	Y= 935.00mm			
Position (Y-Coordinate) (mm)		[y]		935.000
External diameter (mm)		[da]		257.000
Inner diameter (mm)		[di]		110.000
Notch effect			Shoulder	
[D, r, t] (mm)	280.000	1.000	11.500	
Mean roughness (µm)		[Rz]		8.000

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value	[Fzdm, Mbm, Tm, Fqm]	-305128.5	0.0	0.0	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	51101.1	0.0	262295.1
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	-518718.4	86871.9	0.0	445901.7
Cross section, moment of resistance: (mm²)					
	[A, Wb, Wt, A]	42371.4	1610547.7	3221095.4	42371.4

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	-7.201	0.000	0.000	0.000
[σaz, σab, τat, τas]	0.000	31.729	0.000	11.240
[σzmax, σbmax, τtmax, τsmax]	-12.242	53.939	0.000	19.107

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	4.501	4.164	2.486	1.743
References stress slope	[G']	2.448	2.448	1.150	1.150

Support number	[n(r)]	1.209	1.209	1.226	1.226
Support number	[n(d)]	1.004	1.004	1.005	1.005
Mechanical material support factor	[nwm]	1.062	1.062	1.062	1.062

The support factor is determined with the support factor as defined by Stieler.

Notch effect coefficient beta	[Kf]	3.724	3.432	2.017	1.421
Roughness factor	[KR]	0.886	0.886	0.934	0.934
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.200
Design coefficient	[KWK]	3.211	2.967	1.739	1.243
Fatigue limit of part (N/mm²)	[SWK]	93.494	101.185	100.693	140.857

Calculation with individual mean stress:

Mean stress coefficient	[KAK]	1.013	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	94.666	101.185	100.693	140.857
Effective Miner sum	[DM]	1	0.428	1	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SBK]	94.666	101.185	100.693	140.857
Rate of utilization	[aBK]	0.000	0.423	0.000	0.108

Calculation of the combined stress types:
Rate of utilization for the combined load components
a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	7.201
Rate of utilization	[aBKv_1]	0.423

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	7.201
Rate of utilization	[aBKv_2]	0.108

Highest utilization	[aBKmax]	0.423
Safety endurance limit assessment	[S.Dauer]	3.189
Required safety	[jD]	1.350
Result (%)	[S/jD]	236.2

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]	1.270		1.268	
Plastic support number	[npl]	1.0000	1.0570	1.2683	1.0000
Strength of part (N/mm²)	[SSK]	531.63	561.94	389.28	306.94
Rate of utilization	[aSK]	0.032	0.134	0.000	0.087

Rate of utilization for the combined load components:
a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	41.697
Rate of utilization	[aSKvBn]	0.167

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	35.287
Rate of utilization	[aSKvQn]	0.093

Highest utilization [aSKmax] 0.167

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	11.862
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	641.2

Safety against yield point	[S.Rp]	8.402
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	600.2

Cross section 'B-B' Square groove

Comment	Y= 626.00...631.00mm	
Position (Y-Coordinate) (mm)	[y]	628.500
External diameter (mm)	[da]	280.000
Inner diameter (mm)	[di]	180.000
Notch effect		Square groove
[d, r, t, m] (mm)	272.00 0.50 4.00 5.00	
Mean roughness (µm)	[Rz]	8.000

Tension/Compression Bending Torsion Shearing

Load: (N) (Nm)

Mean value [Fzdm, Mbm, Tm, Fqm]	-305128.5	0.0	245216.6	0.0
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	39542.5	0.0	170627.0
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	-518718.4	67222.3	416868.2	290066.0
Cross section, moment of resistance: (mm²) [A, Wb, Wt, A]	32660.0	1596737.7	3193475.3	32660.0

Stresses: (N/mm²)

[σmz, σmb, τmt, τms]	-9.343	0.000	76.787	0.000
[σaz, σab, τat, τas]	0.000	24.765	0.000	10.172
[σzmax, σbmax, τtmax, τsmax]	-15.882	42.100	130.537	17.292

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Notch effect coefficient	[β(dB)]	3.765	3.447	2.500	1.750
[dB] (mm) 280.0, [rB] (mm) 0.5, [r] (mm) 0.5					
Support number	[n(r)]	1.240	1.240	1.260	1.260
Support number	[n(rB)]	1.240	1.240	1.260	1.260
Support number	[n(d)]	1.004	1.004	1.005	1.005
Mechanical material support factor	[nwm]	1.062	1.062	1.062	1.062
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	3.765	3.434	2.488	1.750
Roughness factor	[KR]	1.000	1.000	1.000	1.000
Roughness factor is included into the notch effect coefficient					
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.200
Design coefficient	[KWK]	3.138	2.861	2.073	1.458
Fatigue limit of part (N/mm²)	[SWK]	95.679	104.921	84.475	120.086

Calculation with individual mean stress:

Mean stress coefficient	[KAK]	1.016	1.000	0.915	1.000
Permissible amplitude (N/mm²)	[SAK]	97.199	104.921	77.263	120.086
Effective Miner sum	[DM]	1	0.955	0.3	1
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SBK]	97.199	104.921	77.263	120.086
Rate of utilization	[aBK]	0.000	0.319	0.000	0.114

Calculation of the combined stress types:

Rate of utilization for the combined load components

a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	133.326
Rate of utilization	[aBKv_1]	0.319

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	133.326
Rate of utilization	[aBKv_2]	0.114

Highest utilization	[aBKmax]	0.319
Safety endurance limit assessment	[S.Dauer]	4.237
Required safety	[jD]	1.350
Result (%)	[S/jD]	313.8

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.400

(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.270		1.178
Plastic support number	[npl]	1.0000	1.2700	1.1778	1.0000
Strength of part (N/mm²)	[SSK]	531.63	675.17	361.52	306.94
Rate of utilization	[aSK]	0.042	0.087	0.506	0.079

Rate of utilization for the combined load components:

a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	227.612
Rate of utilization	[aSKvBn]	0.522

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	256.540
Rate of utilization	[aSKvQn]	0.586

Highest utilization [aSKmax] 0.586

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	3.373
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	182.4
Safety against yield point	[S.Rp]	2.390
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	170.7

Cross section 'C-C' Interference fit

Comment	Y= 524.00...622.00mm	
Position (Y-Coordinate) (mm)	[y]	622.000
External diameter (mm)	[da]	280.000
Inner diameter (mm)	[di]	180.000
Notch effect		Interference fit
Characteristics:	Slight interference fit	
Mean roughness (µm)	[Rz]	8.000

	Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)				
Mean value [Fzdm, Mbm, Tm, Fqm]	-305128.5	0.0	245216.6	0.0
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	38433.4	0.0	170643.7
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	-518718.4	65336.8	416868.2	290094.3
Cross section, moment of resistance: (mm²)				
[A, Wb, Wt, A]	36128.3	1787061.3	3574122.6	36128.3

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	-8.446	0.000	68.609	0.000
[σaz, σab, τat, τas]	0.000	21.506	0.000	9.162
[σzmax, σbmax, τtmax, τsmax]	-14.358	36.561	116.635	15.576

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Notch effect coefficient	[β(dB)]	2.401	2.401	1.630	1.315
[dB] (mm) 40.0, [rB] (mm) 2.4, [r] (mm) 16.8					
Support number	[n(r)]	1.070	1.070	1.041	1.041
Support number	[n(rB)]	1.176	1.176	1.141	1.141
Support number	[n(d)]	1.004	1.004	1.005	1.005
Mechanical material support factor	[nwm]	1.062	1.062	1.062	1.062
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.638	2.629	1.778	1.441
Roughness factor	[KR]	1.000	1.000	1.000	1.000
Roughness factor is included into the notch effect coefficient					
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.200
Design coefficient	[KWK]	2.199	2.190	1.482	1.201
Fatigue limit of part (N/mm²)	[SWK]	136.540	137.054	118.182	145.789
Calculation with individual mean stress:					
Mean stress coefficient	[KAK]	1.010	1.000	0.945	1.000
Permissible amplitude (N/mm²)	[SAK]	137.914	137.054	111.738	145.789
Effective Miner sum	[DM]	1	1	0.3	1
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SBK]	137.914	137.054	111.738	145.789
Rate of utilization	[aBK]	0.000	0.212	0.000	0.085

Calculation of the combined stress types:

Rate of utilization for the combined load components

a)For outer surface (shear stress = 0)		
Equivalent mean stress (N/mm²)	[SmV_1]	119.134
Rate of utilization	[aBKv_1]	0.212
b) For neutral line (Bending stress = 0)		
Equivalent mean stress (N/mm²)	[SmV_2]	119.134
Rate of utilization	[aBKv_2]	0.085
Highest utilization	[aBKmax]	0.212
Safety endurance limit assessment	[S.Dauer]	6.373

Required safety	[jD]	1.350
Result (%)	[S/jD]	472.1

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3	[jges]	1.400

(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.270	1.178	
Plastic support number	[npl]	1.0000	1.2700	1.1778	1.0000
Strength of part (N/mm²)	[SSK]	531.63	675.17	361.52	306.94
Rate of utilization	[aSK]	0.038	0.076	0.452	0.071

Rate of utilization for the combined load components:

a)For outer surface (shear stress = 0)		
Equivalent stress (N/mm²)	[SvBn]	203.234
Rate of utilization	[aSKvBn]	0.466
b) For neutral line (Bending stress = 0)		
Equivalent stress (N/mm²)	[SvQn]	229.446
Rate of utilization	[aSKvQn]	0.524
Highest utilization	[aSKmax]	0.524

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	3.771
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	203.9
Safety against yield point	[S.Rp]	2.671
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	190.8

Important remarks concerning strength calculation according to FKM-Guideline:

- Calculation with nominal stresses
- Regulation for proof: Utilization <= 1
- Currently the following restrictions still apply::
  Only for axially symmetrical shafts
- Assumption for calculating the notch factor for shearing:
  ßS = 1.0 + (ßT - 1.0) / 2.0 (according to Prof. Haibach)
- Thread: Determination of notch factor as circumferential groove
- Slight interference fit: determination of the notch factor according to fig. 5.3.11 b) with p = 20MPa
- Proven safety: Effective safety according to special formula,
  condition: safety > required safety or result > 100%

End of Report lines: 1510

A.2.2 KISSsoft Report - Planet shaft

File

Name : Unnamed
Changed by: Joana Mêda de Sousa on: 09.10.2017 at: 23:31:40

Analysis of shafts, axle and beams

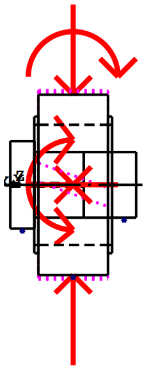
Input data

Coordinate system shaft: see picture W-002

Label		PLGPN
Drawing		
Initial position (mm)		0.000
Length (mm)		463.000
Speed (1/min)		15.00
Sense of rotation: clockwise		
Material		C45 (1)
Young's modulus (N/mm²)		206000.000
Poisson's ratio nu		0.300
Density (kg/m³)		7830.000
Coefficient of thermal expansion	(10^-6/K)	11.500
Temperature (°C)		20.000
Weight of shaft (kg)		188.244
Weight of shaft, including additional masses (kg)		188.244
Mass moment of inertia (kg*m²)		1.671
Momentum of mass GD2 (Nm²)		65.579
Label		PLGSF
Drawing		
Initial position (mm)		88.250
Length (mm)		286.000
Speed (1/min)		24.88
Sense of rotation: counter clockwise		
Material		18CrNiMo7-6
Young's modulus (N/mm²)		206000.000
Poisson's ratio nu		0.300
Density (kg/m³)		7830.000
Coefficient of thermal expansion	(10^-6/K)	11.500
Temperature (°C)		20.000
Weight of shaft (kg)		391.389
Weight of shaft, including additional masses (kg)		391.389
Mass moment of inertia (kg*m²)		30.541
Momentum of mass GD2 (Nm²)		1198.444
Weight towards		(0.000, 0.000, -1.000)
Consider deformations due to shearing		
Shear correction coefficient		1.100
Rolling bearing stiffness is calculated from inner bearing geometry		

Tolerance field: Mean value

Housing material		EN-GJS-400-15 (GGG 40)
Coefficient of thermal expansion	(10^-6/K)	12.500
Temperature of housing (°C)		40.000
Thermal housing reference point (mm)		0.000
Reference temperature (°C)		20.000



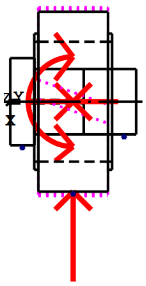


Figure: Load applications

Shaft definition (PLGPN)

Outer contour

Cylinder (Cylinder) 0.000mm ... 89.000mm

Diameter (mm)	[d]	320.0000
Length (mm)	[l]	89.0000
Surface roughness (µm)	[Rz]	8.0000

Chamfer left (Chamfer left)
l=6.00 (mm), alpha=45.00 (°)

Chamfer right (Chamfer right)
l=6.00 (mm), alpha=45.00 (°)

Square groove (Square groove)
b=14.00 (mm), t=5.00 (mm), r=0.50 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder) 89.000mm ... 463.000mm

Diameter (mm)	[d]	240.0000
Length (mm)	[l]	374.0000
Surface roughness (µm)	[Rz]	8.0000

Cross hole (Cross hole)
d=13.00

Inner contour

Cylinder inside (Cylindrical bore) 0.000mm ... 270.000mm

Diameter (mm)	[d]	13.0000
Length (mm)	[l]	270.0000
Surface roughness (µm)	[Rz]	8.0000

Forces

Type of force element		**Coupling**		
Label in the model		PLGPNPCC(PCCCT)		
Position on shaft (mm)	[y_{local}]	231.2500		
Position in global system (mm)	[y_{global}]	231.2500		
Effective diameter (mm)		0.0000		
Radial force factor (-)		0.0000		
Direction of the radial force (°)		0.0000		
Axial force factor (-)		0.0000		
Length of load application (mm)		0.0000		
Power (kW)		0.0000		
Torque (Nm)		-0.0000		
Axial force (load spectrum) (N)		0.0000 /	0.0000 /	0.0000
Shearing force X (load spectrum) (N)		0.0000 /	0.0000 /	0.0000
Shearing force Z (Load spectrum) (N)		0.0000 /	0.0000 /	0.0000
Mass (kg)		0.0000		
Mass moment of inertia Jp (kg*m²)		0.0000		
Mass moment of inertia Jxx (kg*m²)		0.0000		
Mass moment of inertia Jzz (kg*m²)		0.0000		
Eccentricity (mm)		0.0000		

Load spectrum:

No.	Frequency (%)	Speed (1/min)	Power (%)	Torque (%)
1	1.2440e-003	15.000	-61.000	-61.000
2	1.0330e-003	15.000	-38.000	-38.000
3	1.0657e-002	15.000	-23.000	-23.000
4	9.3936e+000	15.000	0.000	0.000
5	2.7248e+001	15.000	23.000	23.000
6	1.9304e+001	15.000	45.000	45.000
7	1.3095e+001	15.000	68.000	68.000
8	2.3136e+001	15.000	98.000	98.000
9	7.7924e+000	15.000	121.000	121.000
10	1.8943e-002	15.000	144.000	144.000
11	5.5000e-005	15.000	167.000	167.000

Bearing

Label in the model		SPGEN
Bearing type		Own Input
Bearing position (mm)	[y_{lokal}]	418.500

Bearing position (mm)	[y_{global}]	418.500
Degrees of freedom		
X: fixedY: freeZ: fixed		
Rx: freeRy: freeRz: free		

Label in the model	SPROT
Bearing type	Own Input

Bearing position (mm)	[y_{lokal}]	44.500
Bearing position (mm)	[y_{global}]	44.500
Degrees of freedom		
X: fixedY: fixedZ: fixed		
Rx: freeRy: freeRz: free		

Shaft definition (PLGSF)

Outer contour

Cylinder (Cylinder) 0.000mm ... 15.000mm

Diameter (mm)	[d]	500.0000
Length (mm)	[l]	15.0000
Surface roughness (µm)	[Rz]	8.0000

Radius right (Radius right)
r=8.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder) 15.000mm ... 271.000mm

Diameter (mm)	[d]	660.0000
Length (mm)	[l]	256.0000
Surface roughness (µm)	[Rz]	8.0000

Cylinder (Cylinder) 271.000mm ... 286.000mm

Diameter (mm)	[d]	500.0000
Length (mm)	[l]	15.0000
Surface roughness (µm)	[Rz]	8.0000

Radius left (Radius left)
r=8.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Inner contour

Cylinder inside (Cylindrical bore) 0.000mm ... 286.000mm

Diameter (mm)	[d]	440.0000
Length (mm)	[l]	286.0000
Surface roughness (µm)	[Rz]	8.0000

Forces

Type of force element		**Cylindrical gear**
Label in the model		PLG(PLRIPGCT)
Position on shaft (mm)	[y_{local}]	143.0000
Position in global system (mm)	[y_{global}]	231.2500
Operating pitch diameter (mm)		682.5294
Helix angle (°)		15.0714 right

Working pressure angle at normal section (°)	20.7167		
Position of contact (°)	-0.0000		
Length of load application (mm)	256.0000		
Power (kW)	406.3211		
Torque (Nm)	-155964.0405		
Axial force (load spectrum) (N)	-75071.7776 /	-46766.0254 /	-28305.7522
Shearing force X (load spectrum) (N)	-109191.3006 /	-68020.8102 /	-41170.4904
Shearing force Z (Load spectrum) (N)	-278780.8426 /	-173666.7544 /	-105114.0882
Bending moment X (Load spectrum) (Nm)	-0.0000 /	-0.0000 /	-0.0000
Bending moment Z (Load spectrum) (Nm)	-25619.3488 /	-15959.5943 /	-9659.7545

Load spectrum, driven (input)

No.	Frequency (%)	Speed (1/min)	Power (kW)	Torque (Nm)
1	1.2440e-003	-24.878	-247.856	95138.065
2	1.0330e-003	-24.878	-154.402	59266.335
3	1.0657e-002	-24.878	-93.454	35871.729
4	9.3936e+000	-24.878	0.000	-0.000
5	2.7248e+001	-24.878	93.454	-35871.729
6	1.9304e+001	-24.878	182.845	-70183.818
7	1.3095e+001	-24.878	276.298	-106055.548
8	2.3136e+001	-24.878	398.195	-152844.760
9	7.7924e+000	-24.878	491.649	-188716.489
10	1.8943e-002	-24.878	585.102	-224588.218
11	5.5000e-005	-24.878	678.556	-260459.948

Type of force element		**Cylindrical gear**		
Label in the model		PLG(SNPLPGCT)		
Position on shaft (mm)	[ylocal]	143.0000		
Position in global system (mm)	[yglobal]	231.2500		
Operating pitch diameter (mm)		692.7164		
Helix angle (°)		15.2859 right		
Working pressure angle at normal section (°)		22.7023		
Position of contact (°)		180.0000		
Length of load application (mm)		256.0000		
Power (kW)		406.3211		
Torque (Nm)		155964.0405		
Axial force (load spectrum) (N)		75071.7776 /	46766.0254 /	28305.7522
Shearing force X (load spectrum) (N)		119129.2476 /	74211.6624 /	44917.5852
Shearing force Z (Load spectrum) (N)		-274681.1255 /	-171112.8323 /	-103568.2932
Bending moment X (Load spectrum) (Nm)		-0.0000 /	-0.0000 /	-0.0000
Bending moment Z (Load spectrum) (Nm)		-26001.7270 /	-16197.7971 /	-9803.9299

Load spectrum, driving (output)

No.	Frequency (%)	Speed (1/min)	Power (kW)	Torque (Nm)
1	1.2440e-003	-24.878	247.856	-95138.065
2	1.0330e-003	-24.878	154.402	-59266.335
3	1.0657e-002	-24.878	93.454	-35871.729
4	9.3936e+000	-24.878	-0.000	0.000
5	2.7248e+001	-24.878	-93.454	35871.729
6	1.9304e+001	-24.878	-182.845	70183.818
7	1.3095e+001	-24.878	-276.298	106055.548
8	2.3136e+001	-24.878	-398.195	152844.760
9	7.7924e+000	-24.878	-491.649	188716.489
10	1.8943e-002	-24.878	-585.102	224588.218
11	5.5000e-005	-24.878	-678.556	260459.948

Bearing

Label in the model PLGSFSP
Bearing type Own Input

Bearing position (mm) [y_{lokal}] 143.000
Bearing position (mm) [y_{global}] 231.250
Degrees of freedom
X: freeY: fixedZ: free
Rx: freeRy: freeRz: free

CONNECTIONS

SKF 32248 J3 (PLGPNBRGEN) 152.500mm
Shaft 'PLGPN' <-> Shaft 'PLGSF'
Set fixed bearing right
d = 240.000 (mm), D = 440.000 (mm), b = 127.000 (mm), r = 0.000 (mm)
C = 2190.000 (kN), C0 = 3100.000 (kN), Cu = 365.000 (kN)
Ctheo = 2188.909 (kN), C0theo = 3100.188 (kN)
Calculation with approximate bearings internal geometry (*)
Z =12, Dpw = 341.122 (mm), Dw = 66.136 (mm)
Lwe = 112.699 (mm), a =113
a =113
Diameter, external race (mm) [d_o] 405.874
Diameter, internal race (mm) [d_i] 276.370
Bearing clearance 0.00 µm
The bearing pressure angle will be considered in the calculation
Position (center of pressure) (mm)
102.5000

SKF 32248 J3 (PLGPNBRROT) 310.000mm
Shaft 'PLGPN' <-> Shaft 'PLGSF'
Set fixed bearing left
d = 240.000 (mm), D = 440.000 (mm), b = 127.000 (mm), r = 0.000 (mm)
C = 2190.000 (kN), C0 = 3100.000 (kN), Cu = 365.000 (kN)
Ctheo = 2188.909 (kN), C0theo = 3100.188 (kN)
Calculation with approximate bearings internal geometry (*)
Z =12, Dpw = 341.122 (mm), Dw = 66.136 (mm)
Lwe = 112.699 (mm), a =113
a =113
Diameter, external race (mm) [d_o] 405.874
Diameter, internal race (mm) [d_i] 276.370
Bearing clearance 0.00 µm
The bearing pressure angle will be considered in the calculation
Position (center of pressure) (mm)
360.0000

Results
Note: the maximum deflection and torsion of the shaft under torque,
the life modification factor aISO, and the bearing's thinnest lubricant film thickness EHL, are predefined for the first load bin.

Shaft

Maximum deflection 280.695 (µm) (PLGSF pos = 88.250 mm)

Mass center of gravity

PLGPN (mm)	207.286
PLGSF (mm)	143.000

Total axial load

PLGPN (N)	0.000
PLGSF (N)	-0.000

Torsion under torque

PLGPN (°)	0.000
PLGSF (°)	0.000

Bearing

Probability of failure	[n]	10.00	%
Axial clearance	[u_A]	10.00	µm
Lubricant	Oil: Castrol Optigear Synthetic X 320		

Lubricant with additive, effect on bearing lifetime confirmed in tests.
Oil lubrication, on-line filtration, ISO4406 -/17/14

Lubricant - service temperature	[T_B]	65.00	°C
Limit for factor aISO	[a_{ISOmax}]	50.00	
Oil level	[h_{oil}]	0.00	mm

Oil injection lubrication

Rolling bearing service life according to ISO/TS 16281:2008

Shaft 'PLGPN' Bearing 'SPGEN'

Position (Y-coordinate)	[y]	418.50	mm

	Bearing reaction force				**Bearing reaction moment**			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	-142.953	0.000	279.054	313.539	0.000	0.000	0.000	0.000
2	-89.066	0.000	174.874	196.249	0.000	0.000	0.000	0.000
3	-53.908	0.000	106.919	119.740	0.000	0.000	0.000	0.000
4	0.000	0.000	2.721	2.721	0.000	0.000	0.000	0.000
5	50.166	0.000	-101.477	113.200	0.000	0.000	0.000	0.000
6	98.135	0.000	-201.132	223.796	0.000	0.000	0.000	0.000
7	148.296	0.000	-305.323	339.432	0.000	0.000	0.000	0.000
8	213.725	0.000	-441.226	490.264	0.000	0.000	0.000	0.000
9	263.887	0.000	-545.419	605.903	0.000	0.000	0.000	0.000
10	314.050	0.000	-649.611	721.542	0.000	0.000	0.000	0.000
11	364.213	0.000	-753.804	837.181	0.000	0.000	0.000	0.000

	Displacement of bearing				**Misalignment of bearing**			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	0.0000	13.0626	0.0000	0.0000	0.085	0.000	0.004	0.085
2	0.0000	12.3740	0.0000	0.0000	0.054	0.000	0.003	0.054
3	0.0000	11.9227	0.0000	0.0000	0.033	0.000	0.002	0.033
4	0.0000	11.5118	0.0000	0.0000	0.001	0.000	0.000	0.001
5	0.0000	11.8836	0.0000	0.0000	-0.031	0.000	-0.001	0.031
6	0.0000	12.5224	0.0000	0.0000	-0.062	0.000	-0.001	0.062
7	0.0000	13.1850	0.0000	0.0000	-0.093	0.000	-0.001	0.093
8	0.0000	14.0446	0.0000	0.0000	-0.134	0.000	-0.001	0.134
9	0.0000	14.7058	0.0000	0.0000	-0.166	0.000	-0.002	0.166

10	0.0000	15.3634	0.0000	0.0000	-0.198	0.000	-0.002	0.198
11	0.0000	16.0208	0.0000	0.0000	-0.229	0.000	-0.002	0.229

Shaft 'PLGPN' Bearing 'SPROT'

Position (Y-coordinate) [y] 44.50 mm

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	133.001	-9.003	280.030	310.009	0.000	0.000	0.000	0.000
2	82.871	-7.335	175.576	194.151	0.000	0.000	0.000	0.000
3	50.158	-6.078	107.438	118.570	0.000	0.000	0.000	0.000
4	-0.000	-14.322	2.965	2.965	0.000	0.000	0.000	0.000
5	-53.909	-4.835	-101.510	114.937	0.000	0.000	0.000	0.000
6	-105.452	-5.390	-201.425	227.359	0.000	0.000	0.000	0.000
7	-159.357	-5.529	-305.897	344.917	0.000	0.000	0.000	0.000
8	-229.669	-5.295	-442.166	498.255	0.000	0.000	0.000	0.000
9	-283.576	-5.116	-546.639	615.816	0.000	0.000	0.000	0.000
10	-337.482	-4.724	-651.112	733.377	0.000	0.000	0.000	0.000
11	-391.390	-4.247	-755.585	850.938	0.000	0.000	0.000	0.000

	Displacement of bearing				Misalignment of bearing			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	0.0000	11.1250	0.0000	0.0000	-0.088	0.000	-0.002	0.088
2	0.0000	11.1250	0.0000	0.0000	-0.055	0.000	-0.001	0.055
3	0.0000	11.1250	0.0000	0.0000	-0.034	0.000	-0.001	0.034
4	0.0000	11.1250	0.0000	0.0000	-0.001	0.000	-0.000	0.001
5	0.0000	11.1250	0.0000	0.0000	0.032	0.000	-0.001	0.032
6	0.0000	11.1250	0.0000	0.0000	0.064	0.000	-0.001	0.064
7	0.0000	11.1250	0.0000	0.0000	0.097	0.000	-0.001	0.097
8	0.0000	11.1250	0.0000	0.0000	0.140	0.000	-0.001	0.140
9	0.0000	11.1250	0.0000	0.0000	0.173	0.000	-0.001	0.173
10	0.0000	11.1250	0.0000	0.0000	0.206	0.000	-0.001	0.206
11	0.0000	11.1250	0.0000	0.0000	0.239	0.000	-0.001	0.239

Shaft 'PLGSF' Bearing 'PLGSFSP'

Position (Y-coordinate) [y] 143.00 mm

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	0.000	9.040	0.000	0.000	0.000	0.000	0.000	0.000
2	0.000	7.347	0.000	0.000	0.000	0.000	0.000	0.000
3	0.000	6.091	0.000	0.000	0.000	0.000	0.000	0.000
4	0.000	14.322	0.000	0.000	0.000	0.000	0.000	0.000
5	0.000	4.847	0.000	0.000	0.000	0.000	0.000	0.000
6	0.000	5.427	0.000	0.000	0.000	0.000	0.000	0.000
7	0.000	5.567	0.000	0.000	0.000	0.000	0.000	0.000
8	0.000	5.333	0.000	0.000	0.000	0.000	0.000	0.000
9	0.000	5.153	0.000	0.000	0.000	0.000	0.000	0.000
10	0.000	4.762	0.000	0.000	0.000	0.000	0.000	0.000
11	0.000	4.285	0.000	0.000	0.000	0.000	0.000	0.000

	Displacement of bearing				Misalignment of bearing			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	-6.5409	57.8125	-98.7716	98.9879	0.082	-0.000	-0.543	0.549
2	-7.0210	57.8125	-69.8332	70.1852	0.086	-0.000	-0.388	0.397

3	-7.2501	57.8125	-48.8563	49.3913	0.089	-0.000	-0.274	0.288
4	0.0000	57.8125	-8.0286	8.0286	0.061	0.000	0.000	0.061
5	9.0435	57.8125	47.1436	48.0032	-0.086	-0.000	0.278	0.291
6	9.5697	57.8125	77.4496	78.0386	-0.081	-0.000	0.441	0.448
7	9.9498	57.8125	105.7349	106.2020	-0.077	-0.000	0.590	0.595
8	10.3718	57.8125	140.0938	140.4772	-0.073	-0.000	0.769	0.773
9	10.5941	57.8125	165.2276	165.5669	-0.069	-0.000	0.900	0.902
10	10.8254	57.8125	189.6128	189.9216	-0.066	-0.000	1.026	1.028
11	11.0567	57.8125	213.5807	213.8667	-0.063	-0.000	1.149	1.151

Rolling bearing 'PLGPNBRGEN'

Position (Y-coordinate)	[y]	152.50	mm
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	2.155	
Nominal bearing service life	[L_{nh}]	93103.72	h
Modified bearing service life	[L_{nmh}]	102759.19	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	3.33	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	0.000	µm
Reference rating service life	[L_{nrh}]	207225.86	h
Modified reference rating service life	[L_{nrmh}]	198251.77	h

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	-194.294	-100.320	-275.887	337.437	13006.968	0.000	-9879.057	16333.309
2	-122.006	-62.281	-172.213	211.052	8114.353	0.000	-6086.448	10143.351
3	-74.423	-37.599	-104.743	128.491	4926.401	0.000	-3643.358	6127.274
4	0.000	-0.459	-1.732	1.732	78.678	0.000	0.000	78.678
5	78.112	-37.231	100.946	127.638	-4755.267	0.000	3821.202	6100.340
6	151.277	-74.076	199.886	250.678	-9438.569	0.000	7593.277	12113.812
7	227.170	-112.974	303.585	379.171	-14331.226	0.000	11580.586	18425.363
8	325.685	-164.089	439.214	546.790	-20686.316	0.000	16808.582	26654.307
9	401.061	-203.408	543.092	675.129	-25538.305	0.000	20844.577	32965.154
10	476.168	-242.844	647.250	803.535	-30400.232	0.000	24890.533	39290.110
11	551.329	-282.405	751.453	932.011	-35242.569	0.000	28933.553	45598.127

	Displacement of bearing				Misalignment of bearing			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	45.4027	-46.8034	88.9239	99.8443	-0.126	0.000	0.520	0.535
2	35.1296	-46.7429	66.3717	75.0952	-0.114	0.000	0.374	0.391
3	27.3163	-46.7052	49.6107	56.6340	-0.106	0.000	0.265	0.286
4	-0.0000	-46.5520	12.6971	12.6971	-0.062	0.000	-0.000	0.062
5	-29.2354	-46.7163	-47.9447	56.1552	0.102	0.000	-0.270	0.289
6	-40.9971	-46.7850	-72.0427	82.8909	0.113	0.000	-0.425	0.440
7	-51.4643	-46.8619	-93.9298	107.1045	0.125	0.000	-0.566	0.580
8	-63.8670	-46.9670	-120.0352	135.9685	0.143	0.000	-0.735	0.749
9	-72.7362	-47.0478	-138.8055	156.7084	0.156	0.000	-0.858	0.872
10	-81.2346	-47.1308	-156.8741	176.6594	0.170	0.000	-0.975	0.990
11	-89.5492	-47.2149	-174.5202	196.1539	0.184	0.000	-1.090	1.106

Rolling bearing 'PLGPNBRROT'

Position (Y-coordinate)	[y]	310.00	mm
Life modification factor for reliability[a_1]		1.000	

Life modification factor	[a_{ISO}]	2.063	
Nominal bearing service life	[L_{nh}]	95166.07	h
Modified bearing service life	[L_{nmh}]	105836.62	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	3.36	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	0.000	µm
Reference rating service life	[L_{nrh}]	237277.81	h
Modified reference rating service life	[L_{nrmh}]	235038.89	h

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	204.246	109.329	-281.360	347.678	-12578.121	0.000	-10342.529	16284.257
2	128.200	69.618	-176.392	218.059	-7786.025	0.000	-6363.774	10055.835
3	78.174	43.679	-107.769	133.137	-4688.822	0.000	-3800.895	6035.881
4	-0.000	14.781	-2.107	2.107	-49.096	0.000	0.000	49.096
5	-74.368	42.068	103.889	127.763	4524.270	0.000	3632.245	5801.915
6	-143.959	79.472	204.523	250.108	9075.582	0.000	7226.328	11601.120
7	-216.108	118.510	309.491	377.475	13868.165	0.000	11040.366	17726.130
8	-309.739	169.391	446.039	543.037	20150.717	0.000	16063.620	25769.968
9	-381.370	208.530	550.830	669.968	24930.769	0.000	19910.951	31905.943
10	-452.733	247.575	655.341	796.517	29764.738	0.000	23790.828	38104.372
11	-524.149	286.659	759.808	923.060	34586.249	0.000	27665.277	44289.685

	Displacement of bearing				Misalignment of bearing			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	-30.7732	-44.5300	75.1287	81.1868	-0.046	0.000	0.522	0.524
2	-20.1207	-45.2984	52.2920	56.0295	-0.063	0.000	0.375	0.380
3	-12.2282	-45.8019	35.2533	37.3138	-0.075	0.000	0.266	0.276
4	-0.0000	-46.2710	3.0520	3.0520	-0.061	0.000	-0.000	0.061
5	11.0094	-45.8443	-34.0900	35.8237	0.073	0.000	-0.270	0.279
6	21.5863	-45.1302	-58.6511	62.4974	0.055	0.000	-0.425	0.428
7	31.1583	-44.3891	-80.8445	86.6411	0.038	0.000	-0.565	0.566
8	42.5466	-43.4272	-107.1669	115.3037	0.017	0.000	-0.733	0.733
9	50.8326	-42.6873	-126.1607	136.0165	0.000	0.000	-0.855	0.855
10	58.7355	-41.9512	-144.3752	155.8655	-0.016	0.000	-0.972	0.972
11	66.4540	-41.2151	-162.1694	175.2571	-0.032	0.000	-1.087	1.087

(*) Note about roller bearings with an approximated bearing geometry:
The internal geometry of these bearings has not been input in the database.
The geometry is back-calculated as specified in ISO 281, from C and C0 (details in the manufacturer's catalog).
For this reason, the geometry may be different from the actual geometry.
This can lead to differences in the service life calculation and, more importantly, the roller bearing stiffness.

Damage (%) [Lreq] (175200.000)

Bin no	B1	B2
1	0.00	0.00
2	0.00	0.00
3	0.00	0.00
4	0.00	0.00
5	0.11	0.08
6	1.27	1.01
7	5.18	4.26
8	44.50	37.36
9	37.12	31.67
10	0.19	0.16

11	0.00	0.00

Σ	88.37	74.54

Utilization (%) [Lreq] (175200.000)

B1	B2
96.36	91.56

Note: Utilization = (Lreq/Lh)^(1/k)

Ball bearing: k = 3, roller bearing: k = 10/3

B1: PLGPNBRGEN (Connecting rolling bearing)

B2: PLGPNBRROT (Connecting rolling bearing)

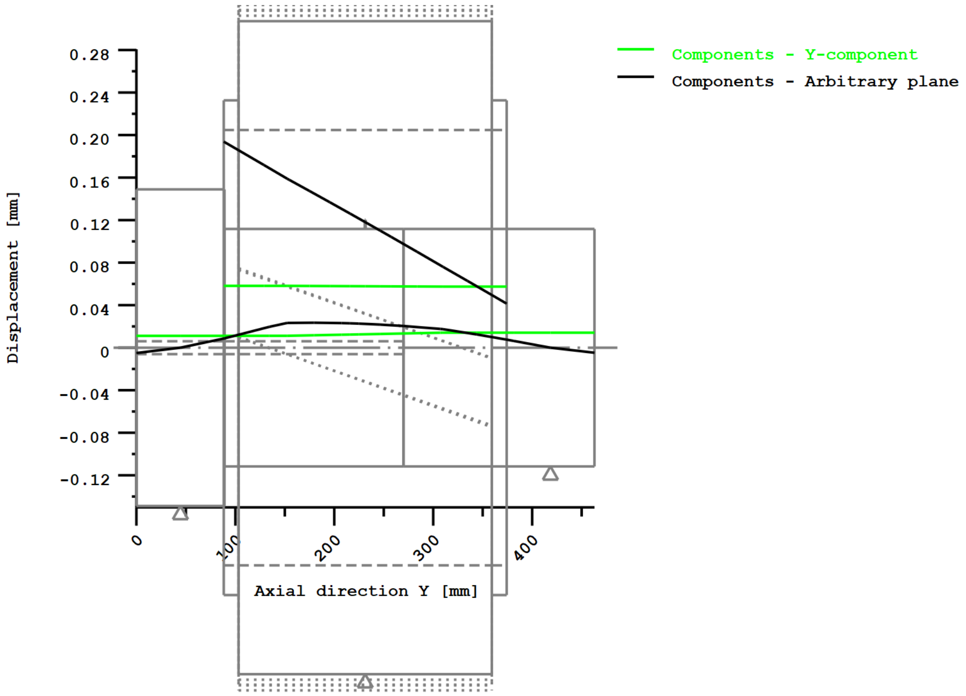
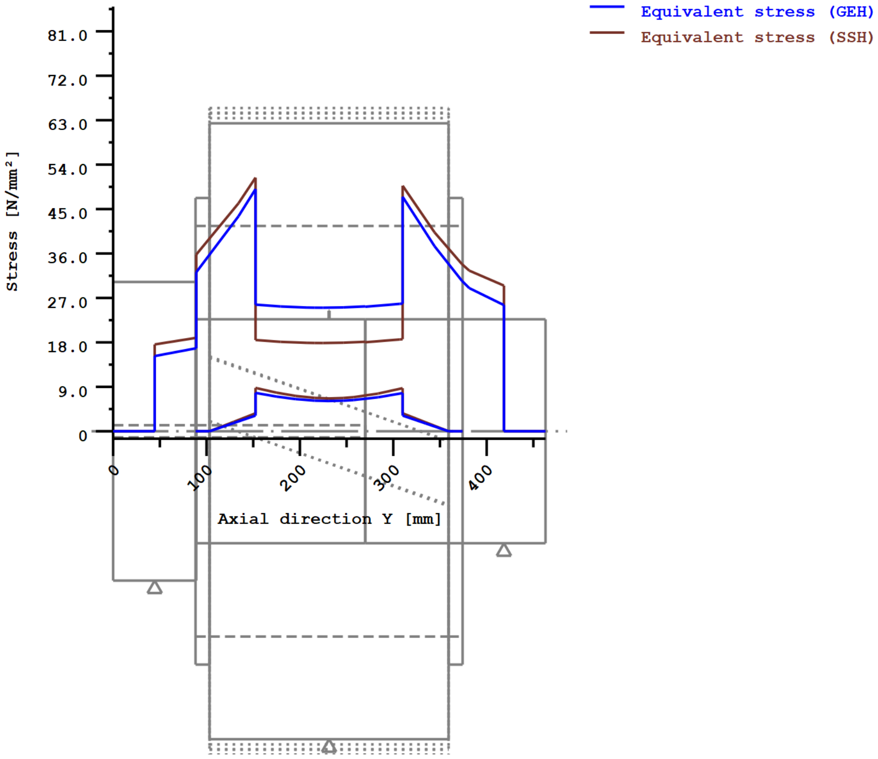


Figure: Deformation (bending etc.) (Arbitrary plane 51.67650271 121)



Nominal stresses, without taking into account stress concentrations
GEH(von Mises): sigV = ((sigB+sigZ,D)^2 + 3*(tauT+tauS)^2)^1/2
SSH(Tresca): sigV = ((sigB-sigZ,D)^2 + 4*(tauT+tauS)^2)^1/2

Figure: Equivalent stress

Strength calculation as specified in the FKM Guideline (6th Edition, 2012)

Summary

PLGSF

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

Calculation of service strength and static strength

S-N curve (Woehler line) according Miner elementary

Rolled steel, case-hardening steel

Overload case F1 (chapter 2.4.2): Constant mean stress

Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 1.5	[jG]	1.00

Cross section	Pos (Y co-ord) (mm)	
A-A	221.75	Smooth shaft
B-B	66.20	Smooth shaft
C-C	127.74	Interference fit

Results:

Cross section	Kfb	KRs	ALGmax	SD	SS	SB
A-A	1.00	0.89	0.02	69.83	60.56	85.50
B-B	1.00	0.89	0.05	28.72	24.73	34.92
C-C	2.71	1.00	0.05	29.43	31.87	44.99
Nominal safety:				1.35	1.40	1.85

Abbreviations:

Kfb: Notch factor bending

KRs: Surface factor

ALGmax: Highest utilization

SD: Safety endurance limit

SS: Safety against yield point

SB: Safety against tensile stress

Service life and damage

System service life (h)	[Hatt]	1000000.00
Damage to system (%)	[D]	0.00

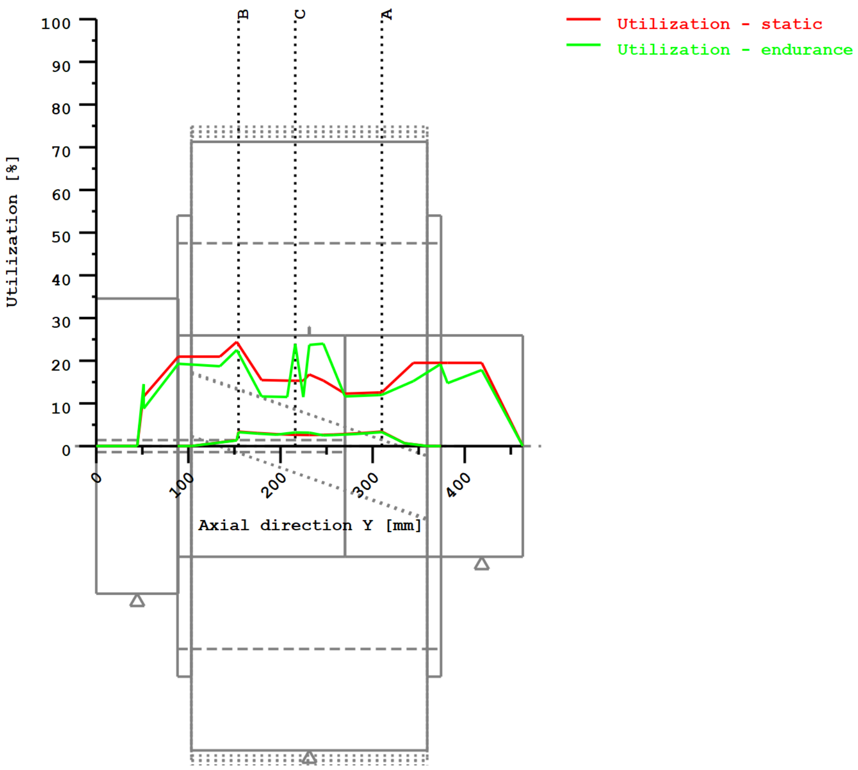
Utilization (%) [Smin/S]

Cross section	Static	Endurance

A-A	2.312	1.933
B-B	5.660	4.701
C-C	4.393	4.587

Maximum utilization of shafts (%)

	[A]
PLGSF:	5.660



Utilization = Smin/S (%)

Figure: Strength

Calculation details

General statements

Label	PLGSF	
Drawing		
Length (mm)	[l]	286.00
Speed (1/min)	[n]	24.88

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

	Tension/Compression	Bending	Torsion	Shearing
Load factor static calculation	1.700	1.700	1.700	1.700
Load factor endurance limit	1.000	1.000	1.000	1.000

Rolled steel, case-hardening steel

Base stress according FKM chapter 5.1:

Tensile strength (N/mm²)	[Rm,N]	1200.00
Yield point (N/mm²)	[Rp,N]	850.00
Fatigue limit (N/mm²)	[σzdWN]	480.00
Fatigue limit (N/mm²)	[σbWN]	510.00

Fatigue limit (N/mm²)	[τtWN]	305.00		
Fatigue limit (N/mm²)	[τsWN]	280.00		
Breaking elongation (%)	[A]	8.00		
Reference diameter (mm)	[deffNm, deffNp]	16.00	16.00	
Required life time	[H]	175200.00		
Number of load cycles	[NL]	261518049		
Service strength for a load spectrum				
S-N curve (Woehler lines) according to Miner elementary according to FKM guideline				
Temperature (°C)	[Temperatur]	20.000		
Temperature duration (h)	[TemperaturD]	175200.000		
Temperature coefficients	[KTm, KTp, KTD]	1.000	1.000	1.000
	[KTtm, KTtp]	1.000	1.000	
Internal stress coefficient	[KEs, KEt]	1.000	1.000	
Additional coefficients	[KA, KW, KfW]	1.000	1.000	1.000
	[KNL, KNLE]	1.000	1.000	
Protective layer factor	[KS]	1.000		

Material properties:

[fσZ, fσD, fτ, Rpmax]	1.000	1.000	0.577	1150.0
[fWt, fWs]	0.577	0.400		
[aM, bM, aTD]	0.35000	-0.100	1.400	
[aG, bG, aRsig, RmNmin]	0.500	2700.0	0.220	400.0
[MS, MT]	0.1627	0.0939		
[kσ, kτ]	15	25		
[kDσ, kDτ]	0	0		
[NDσ, NDτ]	1e+006	1e+006		
[NDσII, NDτII]	0	0		

Thickness of raw material (mm)	[d.eff]	670.00	
Material data calculated acc. FKM directive with Kdm, Kdp			
Geometric size factors (Kdm, Kdp) calculated from raw diameter			
Material strength calculated from size of raw material			
Constants	[adm, adp]	0.370	0.370
Size factors	[Kdm, Kdp]	0.625	0.625
Tensile strength (N/mm²)	[Rm]	750.54	
Yield point (N/mm²)	[Rp]	531.63	
σzdW (N/mm²)	[σzdW]	300.22	
σbW (N/mm²)	[σbW]	318.98	
τtW (N/mm²)	[τtW]	190.76	
τsW (N/mm²)	[τsW]	175.13	

Overload case F1 (chapter 2.4.2): Constant mean stress		
Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jG]	1.00

Cross section 'A-A' Smooth shaft

Comment

Position (Y-Coordinate) (mm) [y] 221.750

External diameter (mm)	[da]	660.000
Inner diameter (mm)	[di]	440.000
Notch effect		Smooth shaft
Mean roughness (µm)	[Rz]	8.000

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value	[Fzdm, Mbm, Tm, Fqm]	-0.0	0.0	0.0	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	27989.9	0.0	290778.6
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	-0.0	47582.9	0.0	494323.6
Cross section, moment of resistance: (mm²)					
[A, Wb, Wt, A]		190066.4	22649574.0	45299148.1	190066.4

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	0.000	10322.088	0.000	107263.552
2	1.0330e-003	0.000	6432.146	0.000	67110.178
3	1.0657e-002	0.000	3895.240	0.000	40923.195
4	9.3936e+000	-0.000	20.599	0.000	769.942
5	2.7248e+001	-0.000	3850.396	0.000	39383.560
6	1.9304e+001	-0.000	7538.362	0.000	77791.135
7	1.3095e+001	-0.000	11393.980	0.000	117944.509
8	2.3136e+001	-0.000	16423.052	0.000	170318.476
9	7.7924e+000	-0.000	20278.677	0.000	210471.851
10	1.8943e-002	-0.000	24134.302	0.000	250625.226
11	5.5000e-005	-0.000	27989.927	0.000	290778.601

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	-0.000	0.000	0.000	0.000
[σaz, σab, τat, τas]	0.000	1.236	0.000	2.981
[σzmax, σbmax, τtmax, τsmax]	-0.000	2.101	0.000	5.068

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	1.000	1.000	1.000	1.000
References stress slope	[G']	0.000	0.000	0.000	0.000
Support number	[n(r)]	1.000	1.000	1.000	1.000
Support number	[n(d)]	1.002	1.002	1.002	1.002
Mechanical material support factor	[nwm]	1.062	1.062	1.062	1.062
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	1.000	0.998	0.998	1.000
Roughness factor	[KR]	0.886	0.886	0.934	0.934
Surface stabilization factor	[KV]	1.000	1.100	1.100	1.100
Design coefficient	[KWK]	1.129	1.025	0.971	0.973
Fatigue limit of part (N/mm²)	[SWK]	265.958	292.968	180.299	179.947

Calculation with individual mean stress:

Mean stress coefficient	[KAK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	265.958	292.968	180.299	179.947
Effective Miner sum	[DM]	1	0.328	0.3	0.3

Coefficient service strength	[KBK]	1.000	1.038	1.000	1.157
Permissible amplitude (N/mm²)	[SBK]	265.958	303.987	180.299	208.183
Rate of utilization	[aBK]	0.000	0.005	0.000	0.019

Calculation of the combined stress types:
Rate of utilization for the combined load components
a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.005

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	0.000
Rate of utilization	[aBKv_2]	0.019

Highest utilization	[aBKmax]	0.019
Safety endurance limit assessment	[S.Dauer]	69.829
Required safety	[jD]	1.350
Result (%)	[S/jD]	5172.5

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.270		1.166
Plastic support number	[npl]	1.0000	1.2700	1.1663	1.0000
Strength of part (N/mm²)	[SSK]	531.63	675.17	357.98	306.94
Rate of utilization	[aSK]	0.000	0.004	0.000	0.023

Rate of utilization for the combined load components:
a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	2.101
Rate of utilization	[aSKvBn]	0.004

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	8.778
Rate of utilization	[aSKvQn]	0.023

Highest utilization	[aSKmax]	0.023

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	85.498
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	4621.5

Safety against yield point	[S.Rp]	60.561
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	4325.8

Cross section 'B-B' Smooth shaft

Comment		
Position (Y-Coordinate) (mm)	[y]	66.200
External diameter (mm)	[da]	660.000
Inner diameter (mm)	[di]	440.000
Notch effect		Smooth shaft
Mean roughness (µm)	[Rz]	8.000

	Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)				
Mean value [Fzdm, Mbm, Tm, Fqm]	-282373.8	0.0	-0.0	0.0
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	29010.9	0.0	707046.8
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	-480035.5	49318.5	0.0	1201979.6
Cross section, moment of resistance: (mm²)				
[A, Wb, Wt, A]	190066.4	22649574.0	45299148.1	190066.4

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	-100288.526	10687.049	-0.000	256078.089
2	1.0330e-003	-62270.777	6662.913	-0.000	160282.595
3	1.0657e-002	-37588.203	4040.984	-0.000	97582.616
4	9.3936e+000	-458.744	59.922	-0.000	933.276
5	2.7248e+001	-37220.368	3906.372	-0.000	97915.517
6	1.9304e+001	-74044.944	7762.709	-0.000	191376.120
7	1.3095e+001	-112943.123	11793.439	-0.000	288773.428
8	2.3136e+001	-164057.594	17026.922	-0.000	415681.734
9	7.7924e+000	-203376.704	21020.535	-0.000	512799.102
10	1.8943e-002	-242812.975	25025.414	-0.000	609886.956
11	5.5000e-005	-282373.847	29010.861	-0.000	707046.848

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	-1.486	0.000	-0.000	0.000
[σaz, σab, τat, τas]	0.000	1.281	0.000	7.249
[σzmax, σbmax, τtmax, τsmax]	-2.526	2.177	0.000	12.324

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	1.000	1.000	1.000	1.000
References stress slope	[G']	0.000	0.000	0.000	0.000
Support number	[n(r)]	1.000	1.000	1.000	1.000
Support number	[n(d)]	1.002	1.002	1.002	1.002
Mechanical material support factor	[nwm]	1.062	1.062	1.062	1.062
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	1.000	0.998	0.998	1.000
Roughness factor	[KR]	0.886	0.886	0.934	0.934
Surface stabilization factor	[KV]	1.000	1.100	1.100	1.100
Design coefficient	[KWK]	1.129	1.025	0.971	0.973
Fatigue limit of part (N/mm²)	[SWK]	265.958	292.968	180.299	179.947
Calculation with individual mean stress:					
Mean stress coefficient	[KAK]	1.001	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	266.200	292.968	180.299	179.947
Effective Miner sum	[DM]	1	0.328	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.038	1.000	1.157
Permissible amplitude (N/mm²)	[SBK]	266.200	303.987	180.299	208.183
Rate of utilization	[aBK]	0.000	0.006	0.000	0.047

Calculation of the combined stress types:
Rate of utilization for the combined load components
a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	1.486
Rate of utilization	[aBKv_1]	0.006

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	1.486
Rate of utilization	[aBKv_2]	0.047

Highest utilization	[aBKmax]	0.047
Safety endurance limit assessment	[S.Dauer]	28.718
Required safety	[jD]	1.350
Result (%)	[S/jD]	2127.3

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.270		1.166
Plastic support number	[npl]	1.0000	1.2700	1.1663	1.0000
Strength of part (N/mm²)	[SSK]	531.63	675.17	357.98	306.94
Rate of utilization	[aSK]	0.007	0.005	0.000	0.056

Rate of utilization for the combined load components:
a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	0.348
Rate of utilization	[aSKvBn]	0.011

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	21.494
Rate of utilization	[aSKvQn]	0.057

Highest utilization	[aSKmax]	0.057

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	34.918
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	1887.5

Safety against yield point	[S.Rp]	24.734
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	1766.7

Cross section 'C-C' Interference fit

Comment Y= 0.75...127.75mm

Position (Y-Coordinate) (mm)	[y]	127.740
External diameter (mm)	[da]	660.000
Inner diameter (mm)	[di]	440.000
Notch effect		Interference fit
Characteristics:	Slight interference fit	
Mean roughness (µm)	[Rz]	8.000

Tension/Compression Bending Torsion Shearing

Load: (N) (Nm)

Mean value [Fzdm, Mbm, Tm, Fqm]	-282373.8	0.0	0.0	0.0
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	45448.4	0.0	546184.3
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	-480035.5	77262.2	0.0	928513.3
Cross section, moment of resistance: (mm²) [A, Wb, Wt, A]	190066.4	22649574.0	45299148.1	190066.4

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	-100288.526	16687.821	0.000	201012.918
2	1.0330e-003	-62270.777	10392.164	0.000	126128.633
3	1.0657e-002	-37588.203	6298.665	0.000	76889.856
4	9.3936e+000	-458.744	89.714	0.000	34.964
5	2.7248e+001	-37220.368	6084.484	0.000	77216.810
6	1.9304e+001	-74044.944	12096.699	0.000	149660.513
7	1.3095e+001	-112943.123	18394.021	0.000	224837.283
8	2.3136e+001	-164057.594	26604.670	0.000	322480.373
9	7.7924e+000	-203376.704	32873.113	0.000	397175.954
10	1.8943e-002	-242812.975	39169.084	0.000	471649.721
11	5.5000e-005	-282373.847	45448.366	0.000	546184.307

Stresses: (N/mm²)

[σmz, σmb, τmt, τms]	-1.486	0.000	0.000	0.000
[σaz, σab, τat, τas]	0.000	2.007	0.000	5.600
[σzmax, σbmax, τtmax, τsmax]	-2.526	3.411	0.000	9.520

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Notch effect coefficient	[β(dB)]	2.401	2.401	1.630	1.315
[dB] (mm) 40.0, [rB] (mm) 2.4, [r] (mm) 39.6					
Support number	[n(r)]	1.040	1.040	1.017	1.017
Support number	[n(rB)]	1.176	1.176	1.141	1.141
Support number	[n(d)]	1.002	1.002	1.002	1.002
Mechanical material support factor	[nwm]	1.062	1.062	1.062	1.062
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.716	2.712	1.825	1.475
Roughness factor	[KR]	1.000	1.000	1.000	1.000
Roughness factor is included into the notch effect coefficient					
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.200
Design coefficient	[KWK]	2.263	2.260	1.521	1.229
Fatigue limit of part (N/mm²)	[SWK]	132.648	132.860	115.167	142.472

Calculation with individual mean stress:

Mean stress coefficient	[KAK]	1.002	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	132.890	132.860	115.167	142.472
Effective Miner sum	[DM]	1	0.325	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.039	1.000	1.157
Permissible amplitude (N/mm²)	[SBK]	132.890	138.051	115.167	164.828
Rate of utilization	[aBK]	0.000	0.020	0.000	0.046

Calculation of the combined stress types:

Rate of utilization for the combined load components

a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	1.486
Rate of utilization	[aBKv_1]	0.020
b) For neutral line (Bending stress = 0)		
Equivalent mean stress (N/mm²)	[SmV_2]	1.486
Rate of utilization	[aBKv_2]	0.046
Highest utilization	[aBKmax]	0.046
Safety endurance limit assessment	[S.Dauer]	29.434
Required safety	[jD]	1.350
Result (%)	[S/jD]	2180.3

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.400

(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]	1.270		1.166	
Plastic support number	[npl]	1.0000	1.2700	1.1663	1.0000
Strength of part (N/mm²)	[SSK]	531.63	675.17	357.98	306.94
Rate of utilization	[aSK]	0.007	0.007	0.000	0.043

Rate of utilization for the combined load components:

a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	0.886
Rate of utilization	[aSKvBn]	0.014
b) For neutral line (Bending stress = 0)		
Equivalent stress (N/mm²)	[SvQn]	16.681
Rate of utilization	[aSKvQn]	0.044
Highest utilization	[aSKmax]	0.044

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	44.993
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	2432.1
Safety against yield point	[S.Rp]	31.870
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	2276.4

Important remarks concerning strength calculation according to FKM-Guideline:

- Calculation with nominal stresses
- Regulation for proof: Utilization <= 1
- Currently the following restrictions still apply::
  Only for axially symmetrical shafts
- Assumption for calculating the notch factor for shearing:
  ßS = 1.0 + (ßT - 1.0) / 2.0 (according to Prof. Haibach)
- Thread: Determination of notch factor as circumferential groove
- Slight interference fit: determination of the notch factor according to fig. 5.3.11 b) with p = 20MPa
- Proven safety: Effective safety according to special formula,
  condition: safety > required safety or result > 100%

End of Report	lines:	1053

A.2.3 KISSsoft Report - Intermediate shaft (superior branch)

KISSsoft Release 03/2017 A

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File

Name : Unnamed

Changed by: Joana Mêda de Sousa on: 09.10.2017 at: 22:02:57

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	IMSFSUP
Drawing	
Initial position (mm)	0.000
Length (mm)	881.500
Speed (1/min)	373.85
Sense of rotation: counter clockwise	
Material	18CrNiMo7-6
Young's modulus (N/mm²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m³)	7830.000
Coefficient of thermal expansion (10^-6/K)	11.500
Temperature (°C)	40.000
Weight of shaft (kg)	180.906
Weight of shaft, including additional masses (kg)	180.906
Mass moment of inertia (kg*m²)	0.811
Momentum of mass GD2 (Nm²)	31.823
Weight towards	(0.000, 0.000, -1.000)
Consider deformations due to shearing	
Shear correction coefficient	1.100
Rolling bearing stiffness is calculated from inner bearing geometry	
Tolerance field: Mean value	
Reference temperature (°C)	40.000

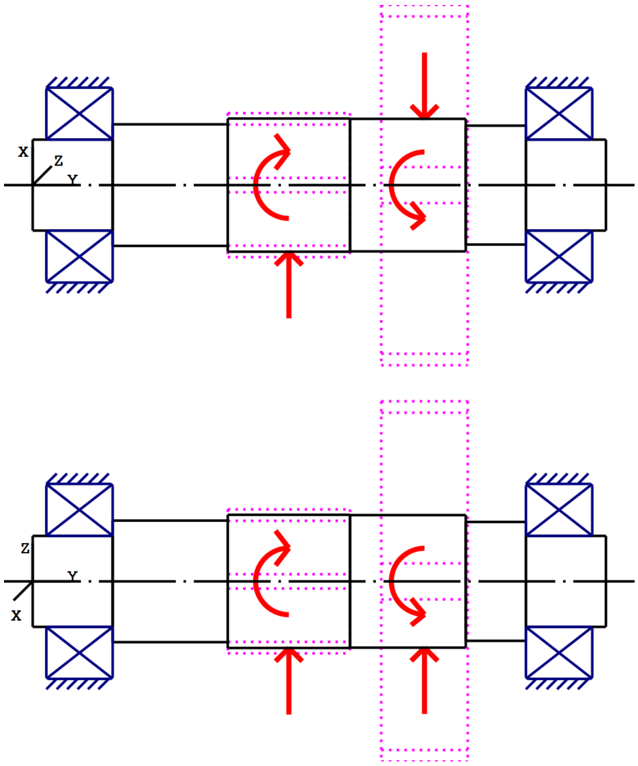


Figure: Load applications

Shaft definition (IMSFSUP)

Outer contour

Cylinder (Cylinder) 0.000mm ... 123.000mm

Diameter (mm)	[d]	140.0000
Length (mm)	[l]	123.0000
Surface roughness (µm)	[Rz]	8.0000

Relief groove right (Relief groove left)
r=1.20 (mm), t=0.40 (mm), l=4.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)
Form F (DIN 509), Series 1, with the usual stressing

Chamfer left (Chamfer left)
l=10.00 (mm), alpha=45.00 (°)

Square groove (Square groove)
b=4.00 (mm), t=3.00 (mm), r=0.50 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder) 123.000mm ... 300.000mm

Diameter (mm)	[d]	187.0000
Length (mm)	[l]	177.0000
Surface roughness (µm)	[Rz]	8.0000

Chamfer left (Chamfer left)
l=10.00 (mm), alpha=45.00 (°)

Radius right (Radius right)

r=6.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder)			300.000mm ...	488.000mm
Diameter (mm)	[d]	205.0000		
Length (mm)	[l]	188.0000		
Surface roughness (µm)	[Rz]	8.0000		

Cylinder (Cylinder)			488.000mm ...	666.000mm
Diameter (mm)	[d]	204.0000		
Length (mm)	[l]	178.0000		
Surface roughness (µm)	[Rz]	8.0000		

Spline (Spline) 439.500mm ... 590.500mm

da=204.00 (mm), df=196.50 (mm), z=67, mn=3.00 (mm), l=151.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Chamfer right (Chamfer right)

l=3.00 (mm), alpha=45.00 (°)

Radius left (Radius left)

r=0.50 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder)			666.000mm ...	758.500mm
Diameter (mm)	[d]	183.0000		
Length (mm)	[l]	92.5000		
Surface roughness (µm)	[Rz]	8.0000		

Radius left (Radius left)

r=5.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Chamfer right (Chamfer right)

l=8.00 (mm), alpha=45.00 (°)

Cylinder (Cylinder)			758.500mm ...	881.500mm
Diameter (mm)	[d]	140.0000		
Length (mm)	[l]	123.0000		
Surface roughness (µm)	[Rz]	8.0000		

Relief groove left (Relief groove right)

r=1.20 (mm), t=0.40 (mm), l=4.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Form F (DIN 509), Series 1, with the usual stressing

Square groove (Square groove)

b=4.00 (mm), t=3.00 (mm), r=0.50 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Chamfer right (Chamfer right)

l=10.00 (mm), alpha=45.00 (°)

Forces

Type of force element		**Cylindrical gear**
Label in the model		IMGSUPGEN(IMHSSUPPGCT)
Position on shaft (mm)	[y_{local}]	602.5000

Position in global system (mm)	$[y_{global}]$	602.5000		
Operating pitch diameter (mm)		554.4000		
Helix angle (°)		25.1077 Double helical gearing, right-left		
Working pressure angle at normal section (°)		20.6227		
Position of contact (°)		-112.0085		
Length of load application (mm)		133.0000		
Power (kW)		1210.0004		
Torque (Nm)		30907.5080		
Axial force (load spectrum) (N)		0.0000 /	0.0000 /	0.0000
Shearing force X (load spectrum) (N)		73650.7723 /	45880.8090 /	27769.9633
Shearing force Z (Load spectrum) (N)		718.8059 /	447.7807 /	271.0252
Bending moment X (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000
Bending moment Z (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000

Load spectrum, driving (output)

No.	Frequency (%)	Speed (1/min)	Power (kW)	Torque (Nm)
1	1.2440e-003	-373.846	738.100	-18853.580
2	1.0330e-003	-373.846	459.800	-11744.853
3	1.0657e-002	-373.846	278.300	-7108.727
4	9.3936e+000	-373.846	-0.000	0.000
5	2.7248e+001	-373.846	-278.300	7108.727
6	1.9304e+001	-373.846	-544.500	13908.379
7	1.3095e+001	-373.846	-822.800	21017.105
8	2.3136e+001	-373.846	-1185.800	30289.358
9	7.7924e+000	-373.846	-1464.101	37398.085
10	1.8943e-002	-373.846	-1742.401	44506.812
11	5.5000e-005	-373.846	-2020.701	51615.538

Type of force element		**Cylindrical gear**		
Label in the model		IMGSUPROT(LSIMSUPPGCT)		
Position on shaft (mm)	$[y_{local}]$	394.0000		
Position in global system (mm)	$[y_{global}]$	394.0000		
Operating pitch diameter (mm)		221.5517		
Helix angle (°)		25.0871 Double helical gearing, right-left		
Working pressure angle at normal section (°)		20.5050		
Position of contact (°)		-30.0000		
Length of load application (mm)		188.0000		
Power (kW)		1210.0004		
Torque (Nm)		-30907.5080		
Axial force (load spectrum) (N)		0.0000 /	0.0000 /	0.0000
Shearing force X (load spectrum) (N)		-145962.6767 /	-90927.5691 /	-55035.1076
Shearing force Z (Load spectrum) (N)		-112253.5208 /	-69928.4228 /	-42325.0980
Bending moment X (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000
Bending moment Z (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000

Load spectrum, driven (input)

No.	Frequency (%)	Speed (1/min)	Power (kW)	Torque (Nm)
1	1.2440e-003	-373.846	-738.100	18853.580
2	1.0330e-003	-373.846	-459.800	11744.853
3	1.0657e-002	-373.846	-278.300	7108.727
4	9.3936e+000	-373.846	0.000	-0.000
5	2.7248e+001	-373.846	278.300	-7108.727
6	1.9304e+001	-373.846	544.500	-13908.379
7	1.3095e+001	-373.846	822.800	-21017.105
8	2.3136e+001	-373.846	1185.800	-30289.358
9	7.7924e+000	-373.846	1464.101	-37398.085
10	1.8943e-002	-373.846	1742.401	-44506.812

11	5.5000e-005	-373.846	2020.701	-51615.538

Bearing

Label in the model		IMSFBRSUPGEN
Bearing type		SKF NCF 2328 ECJB
Bearing type		Cylindrical roller bearing (single row)
		SKF Explorer
Bearing position (mm)	[y_{lokal}]	809.500
Bearing position (mm)	[y_{global}]	809.500
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	140.000
External diameter (mm)	[D]	300.000
Width (mm)	[b]	102.000
Corner radius (mm)	[r]	4.000
Number of rolling bodies	[Z]	11
Rolling body reference circle (mm)	[D_{pw}]	212.652
Diameter rolling body (mm)	[D_w]	54.567
Rolling body length (mm)	[L_{we}]	77.914
Diameter, external race (mm)	[d_o]	267.260
Diameter, internal race (mm)	[d_i]	158.044
Calculation with approximate bearings internal geometry (*)		
Bearing clearance		DIN 620:1988 C0 (82.50 µm)
Basic static load rating (kN)	[C_0]	1530.000
Basic dynamic load rating (kN)	[C]	1250.000
Fatigue load rating (kN)	[C_u]	163.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	1249.750
Basic static load rating (kN)	[C_{0theo}]	1529.726

Label in the model		IMSFBRSUPROT
Bearing type		SKF NCF 2328 ECJB
Bearing type		Cylindrical roller bearing (single row)
		SKF Explorer
Bearing position (mm)	[y_{lokal}]	72.000
Bearing position (mm)	[y_{global}]	72.000
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	140.000
External diameter (mm)	[D]	300.000
Width (mm)	[b]	102.000
Corner radius (mm)	[r]	4.000
Number of rolling bodies	[Z]	11
Rolling body reference circle (mm)	[D_{pw}]	212.652
Diameter rolling body (mm)	[D_w]	54.567
Rolling body length (mm)	[L_{we}]	77.914
Diameter, external race (mm)	[d_o]	267.260
Diameter, internal race (mm)	[d_i]	158.044
Calculation with approximate bearings internal geometry (*)		
Bearing clearance		DIN 620:1988 C0 (82.50 µm)
Basic static load rating (kN)	[C_0]	1530.000
Basic dynamic load rating (kN)	[C]	1250.000
Fatigue load rating (kN)	[C_u]	163.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	1249.750
Basic static load rating (kN)	[C_{0theo}]	1529.726

Results

Note: the maximum deflection and torsion of the shaft under torque,
the life modification factor aISO, and the bearing's thinnest lubricant film thickness EHL, are predefined for the first load bin.

Shaft

Maximum deflection (µm)	499.781
Position of the maximum (mm)	425.333
Mass center of gravity (mm)	442.858
Total axial load (N)	0.000
Torsion under torque (°)	0.027

Bearing

Probability of failure	[n]	10.00	%
Axial clearance	[u_A]	10.00	µm
Lubricant	Oil: Castrol Optigear Synthetic X 320		

Lubricant with additive, effect on bearing lifetime confirmed in tests.
Oil lubrication, on-line filtration, ISO4406 -/17/14

Lubricant - service temperature	[T_B]	65.00	°C
Limit for factor aISO	[a_{ISOmax}]	50.00	
Oil level	[h_{oil}]	0.00	mm

Oil injection lubrication

Rolling bearing service life according to ISO/TS 16281:2008

Shaft 'IMSFSUP' Rolling bearing 'IMSFBRSUPGEN'

Position (Y-coordinate)	[y]	809.50	mm
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	50.000	
Nominal bearing service life	[L_{nh}]	51037.56	h
Modified bearing service life	[L_{nmh}]	328079.54	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	4.67	

Calculation with approximate bearings internal geometry

Operating bearing clearance	[Pd]	77.079	µm
Reference rating service life	[L_{nrh}]	103559.48	h
Modified reference rating service life	[L_{nrmh}]	475979.83	h

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	10.581	0.000	49.305	50.428	-142.542	0.000	48.569	150.589
2	6.624	0.000	31.060	31.759	-63.345	0.000	22.467	67.211
3	4.027	0.000	19.162	19.581	-26.017	0.000	9.466	27.686
4	0.000	0.000	0.892	0.892	-0.112	0.000	0.001	0.112
5	10.258	0.000	-43.175	44.377	94.067	0.000	17.483	95.678
6	20.087	0.000	-85.320	87.653	290.981	0.000	52.656	295.707
7	30.380	0.000	-129.374	132.894	556.408	0.000	101.837	565.651
8	43.825	0.000	-186.828	191.899	951.864	0.000	176.391	968.070

9	54.129	0.000	-230.855	237.116	1287.957	0.000	234.793	1309.183
10	64.448	0.000	-274.894	282.348	1639.741	0.000	298.916	1666.764
11	74.759	0.000	-318.941	327.586	2011.515	0.000	359.897	2043.457

	Displacement of bearing				Misalignment of bearing			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	-17.3944	0.0000	-74.5918	76.5931	0.219	-0.289	-0.212	0.305
2	-14.4560	0.0000	-66.0894	67.6519	0.134	-0.180	-0.149	0.200
3	-12.1543	0.0000	-59.6129	60.8393	0.079	-0.109	-0.105	0.131
4	2.9443	0.0000	-43.1143	43.2147	0.006	0.000	0.001	0.006
5	-18.9353	0.0000	71.5942	74.0559	-0.186	0.109	0.005	0.186
6	-22.4522	0.0000	88.8265	91.6201	-0.355	0.213	-0.012	0.355
7	-24.6180	0.0000	104.3897	107.2533	-0.530	0.322	-0.032	0.531
8	-27.6811	0.0000	122.5561	125.6433	-0.756	0.465	-0.059	0.759
9	-30.4850	0.0000	135.3038	138.6956	-0.929	0.574	-0.078	0.932
10	-33.2377	0.0000	147.4884	151.1873	-1.101	0.683	-0.096	1.106
11	-36.0129	0.0000	159.2185	163.2405	-1.273	0.792	-0.114	1.278

Shaft 'IMSFSUP' Rolling bearing 'IMSFBRSUPROT'

Position (Y-coordinate)	[y]	72.00	mm
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	50.000	
Nominal bearing service life	[L_{nh}]	54880.95	h
Modified bearing service life	[L_{nmh}]	360113.11	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	4.77	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	82.500	µm
Reference rating service life	[L_{nrh}]	109726.18	h
Modified reference rating service life	[L_{nrmh}]	516144.57	h

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	61.713	0.000	63.979	88.892	194.513	0.000	-167.182	256.486
2	38.410	0.000	40.175	55.582	86.791	0.000	-72.082	112.821
3	23.235	0.000	24.661	33.883	35.128	0.000	-27.769	44.778
4	-0.000	0.000	0.882	0.882	-0.019	0.000	-0.000	0.019
5	0.386	0.000	-43.359	43.360	-93.214	0.000	-4.267	93.311
6	0.739	0.000	-85.686	85.689	-295.387	0.000	-13.774	295.708
7	1.090	0.000	-129.943	129.948	-570.907	0.000	-24.032	571.413
8	1.530	0.000	-187.679	187.686	-986.172	0.000	-33.567	986.743
9	1.866	0.000	-231.916	231.923	-1334.624	0.000	-42.791	1335.310
10	2.191	0.000	-276.181	276.190	-1704.889	0.000	-48.359	1705.575
11	2.524	0.000	-320.439	320.449	-2089.808	0.000	-56.936	2090.584

	Displacement of bearing				Misalignment of bearing			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	-67.3491	-0.0000	-67.4440	95.3132	-0.267	0.000	0.153	0.308
2	-57.6354	-0.0000	-58.2860	81.9700	-0.173	0.000	0.080	0.191
3	-49.4772	-0.0000	-51.9891	71.7695	-0.110	0.000	0.034	0.115
4	3.3583	-0.0000	-45.8852	46.0080	0.001	-0.000	0.001	0.001
5	1.3335	-0.0000	76.7540	76.7656	0.178	-0.000	0.037	0.182
6	0.9098	-0.0000	93.9346	93.9390	0.354	0.000	0.050	0.358
7	-0.2495	-0.0000	109.1999	109.2002	0.537	0.000	0.060	0.540
8	-1.9548	-0.0000	127.2311	127.2461	0.772	-0.000	0.074	0.776

9	-2.9797	-0.0000	140.1336	140.1653	0.951	0.000	0.085	0.955
10	-3.5367	-0.0000	152.4159	152.4570	1.130	-0.000	0.097	1.134
11	-3.8465	-0.0000	164.2884	164.3334	1.308	-0.000	0.110	1.312

(*) Note about roller bearings with an approximated bearing geometry:
The internal geometry of these bearings has not been input in the database.
The geometry is back-calculated as specified in ISO 281, from C and C0 (details in the manufacturer's catalog).
For this reason, the geometry may be different from the actual geometry.
This can lead to differences in the service life calculation and, more importantly, the roller bearing stiffness.

Damage (%) [Lreq] (175200.000)

Bin no	B1	B2
1	0.00	0.00
2	0.00	0.00
3	0.00	0.00
4	0.00	0.00
5	0.02	0.02
6	0.26	0.24
7	1.54	1.40
8	17.70	16.23
9	17.20	15.96
10	0.10	0.09
11	0.00	0.00
Σ	36.81	33.94

Utilization (%) [Lreq] (175200.000)

B1	B2
74.09	72.32

Note: Utilization = (Lreq/Lh)^(1/k)
Ball bearing: k = 3, roller bearing: k = 10/3

B1: IMSFBRSUPGEN
B2: IMSFBRSUPROT

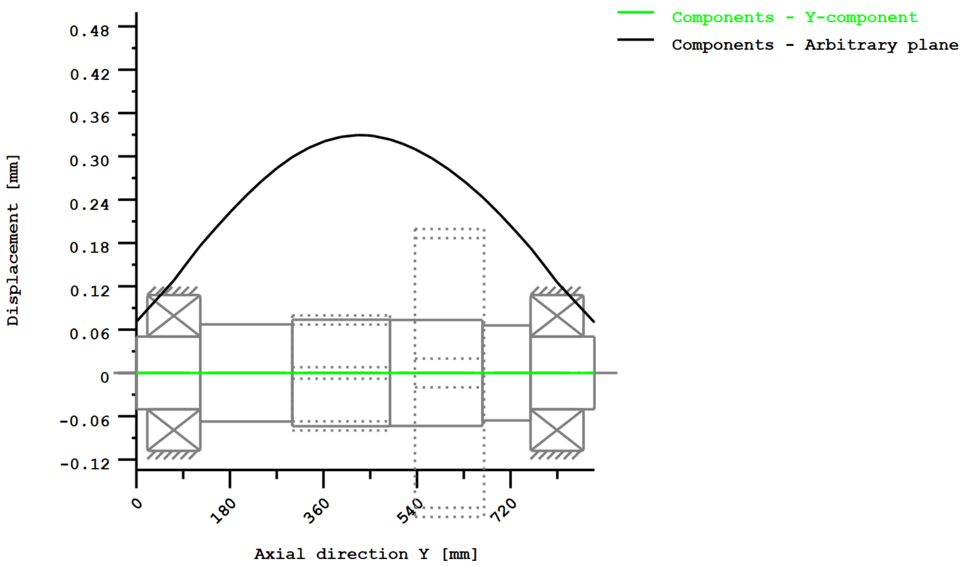
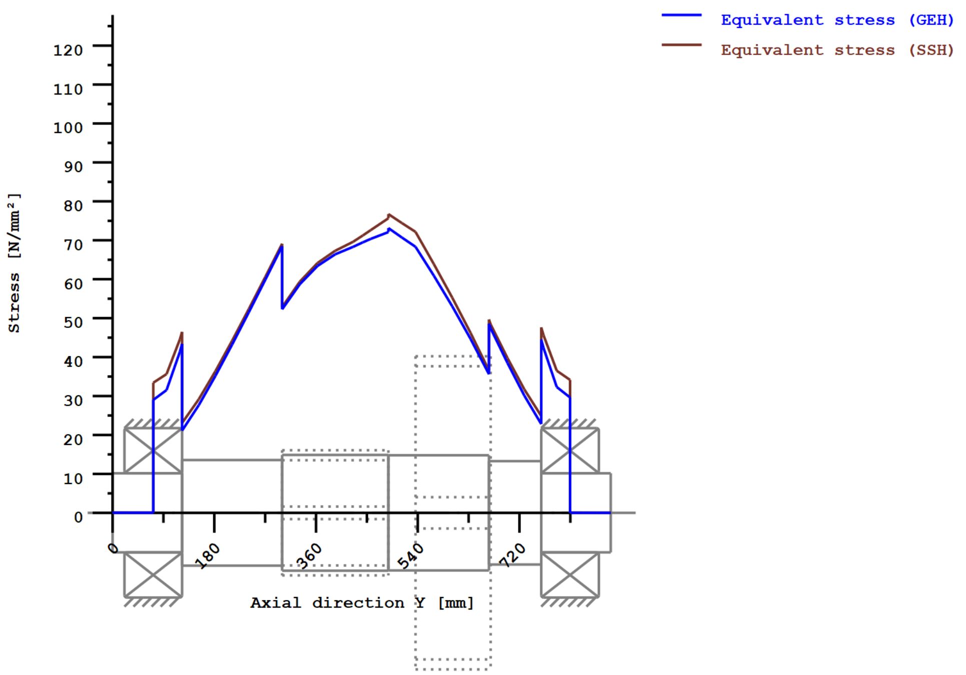


Figure: Deformation (bending etc.) (Arbitrary plane 94.99318738 121)



Nominal stresses, without taking into account stress concentrations
GEH(von Mises): sigV = ((sigB+sigZ,D)^2 + 3*(tauT+tauS)^2)^1/2
SSH(Tresca): sigV = ((sigB-sigZ,D)^2 + 4*(tauT+tauS)^2)^1/2

Figure: Equivalent stress

Strength calculation as specified in the FKM Guideline (6th Edition, 2012)

Summary

IMSFSUP

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

Calculation of service strength and static strength

S-N curve (Woehler line) according Miner elementary

Rolled steel, case-hardening steel

Overload case F1 (chapter 2.4.2): Constant mean stress

Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 1.5	[jG]	1.00

Cross section	Pos (Y co-ord) (mm)	
A-A	488.00	Shoulder
B-B	300.00	Shoulder
C-C	666.00	Shoulder
D-D	123.00	Shoulder with relief groove

Results:

Cross section	Kfb	KRs	ALGmax	SD	SS	SB
A-A	1.66	0.88	0.69	1.96	4.00	5.65
B-B	1.85	0.88	0.79	1.70	4.23	5.98
C-C	1.98	0.88	0.57	2.36	6.26	8.84
D-D	3.10	0.88	0.63	2.15	6.67	9.42
Nominal safety:				1.35	1.40	1.85

Abbreviations:

Kfb: Notch factor bending

KRs: Surface factor

ALGmax: Highest utilization

SD: Safety endurance limit

SS: Safety against yield point

SB: Safety against tensile stress

Service life and damage

System service life (h)	[Hatt]	1000000.00
Damage to system (%)	[D]	0.00
Damage (%)	[H] (175200.0 h)	

Calculation of reliability R(t) using a Weibull distribution; t in (h):

R(t) = 100 * Exp(-((t*fac - t0)/(T - t0))^b) %

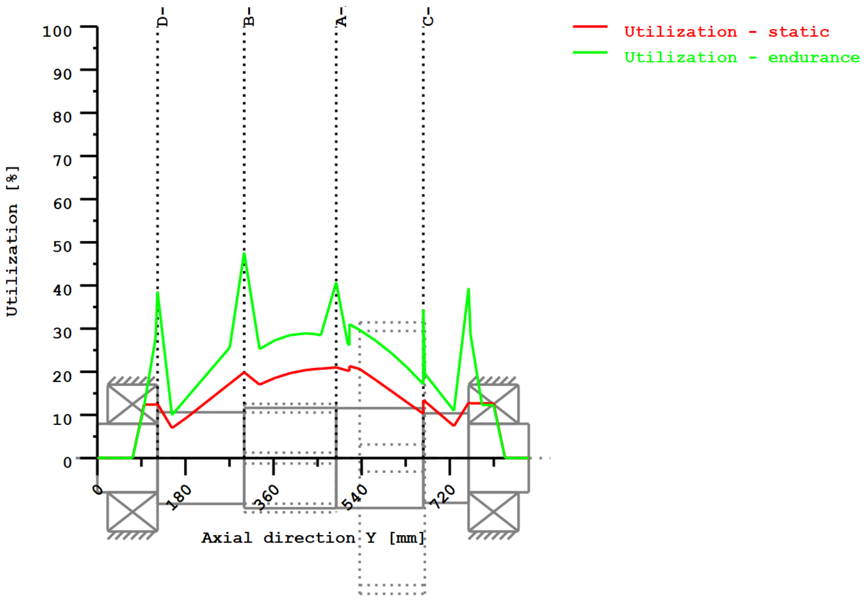
Welle	fac	b	t0	T
1	22431	1.5	2.045e+010	4.338e+010

Damage to cross sections (%)	[D]
A-A:	0.00
B-B:	0.00
C-C:	0.00
D-D:	0.00

Utilization (%) [Smin/S]

Cross section	Static	Endurance
A-A	35.009	69.030
B-B	33.060	79.213
C-C	22.348	57.130
D-D	20.983	62.814

Maximum utilization (%)	[A]	79.213



Utilization = Smin/S (%)

Figure: Strength

Calculation details

General statements

Label	IMSFSUP		
Drawing			
Length (mm)		[l]	881.50
Speed (1/min)		[n]	373.85

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

	Tension/Compression	Bending	Torsion	Shearing
Load factor static calculation	1.700	1.700	1.700	1.700
Load factor endurance limit	1.000	1.000	1.000	1.000

Rolled steel, case-hardening steel

Base stress according FKM chapter 5.1:

Tensile strength (N/mm²)	[Rm,N]	1200.00	
Yield point (N/mm²)	[Rp,N]	850.00	
Fatigue limit (N/mm²)	[σzdWN]	480.00	
Fatigue limit (N/mm²)	[σbWN]	510.00	
Fatigue limit (N/mm²)	[τtWN]	305.00	
Fatigue limit (N/mm²)	[τsWN]	280.00	
Breaking elongation (%)	[A]	8.00	
Reference diameter (mm)	[deffNm, deffNp]	16.00	16.00

Required life time	[H]	175200.00
Number of load cycles	[NL]	3929870769

Service strength for a load spectrum

S-N curve (Woehler lines) according to Miner elementary according to FKM guideline

Temperature (°C)	[Temperatur]	40.000

Temperature duration (h)	[TemperaturD]	175200.000		
Temperature coefficients	[KTm, KTp, KTD]	1.000	1.000	1.000
	[KTtm, KTtp]	1.000	1.000	
Internal stress coefficient	[KEs, KEt]	1.000	1.000	
Additional coefficients	[KA, KW, KfW]	1.000	1.000	1.000
	[KNL, KNLE]	1.000	1.000	
Protective layer factor	[KS]	1.000		

Material properties:

[fσZ, fσD, fτ, Rpmax]	1.000	1.000	0.577	1150.0
[fWt, fWs]	0.577	0.400		
[aM, bM, aTD]	0.35000	-0.100	1.400	
[aG, bG, aRsig, RmNmin]	0.500	2700.0	0.220	400.0
[MS, MT]	0.1727	0.0997		
[kσ, kτ]	15	25		
[kDσ, kDτ]	0	0		
[NDσ, NDτ]	1e+006	1e+006		
[NDσII, NDτII]	0	0		

Thickness of raw material (mm)	[d.eff]	210.00	
Material data calculated acc. FKM directive with Kdm, Kdp			
Geometric size factors (Kdm, Kdp) calculated from raw diameter			
Material strength calculated from size of raw material			
Constants	[adm, adp]	0.370	0.370
Size factors	[Kdm, Kdp]	0.649	0.649
Tensile strength (N/mm²)	[Rm]	779.05	
Yield point (N/mm²)	[Rp]	551.83	
σzdW (N/mm²)	[σzdW]	311.62	
σbW (N/mm²)	[σbW]	331.10	
τtW (N/mm²)	[τtW]	198.01	
τsW (N/mm²)	[τsW]	181.78	

Overload case F1 (chapter 2.4.2): Constant mean stress

Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jG]	1.00

Cross section 'A-A' Shoulder

Comment	Y= 488.00mm			
Position (Y-Coordinate) (mm)		[y]		488.000
External diameter (mm)		[da]		204.000
Inner diameter (mm)		[di]		0.000
Notch effect			Shoulder	
[D, r, t] (mm)	205.000	0.500	0.500	
Mean roughness (µm)		[Rz]		8.000

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value [Fzdm, Mbm, Tm, Fqm]		-0.0	0.0	51615.5	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	84762.2	0.0	191042.9
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	-0.0144095.8		87746.4	324772.9
Cross section, moment of resistance: (mm²)					
[A, Wb, Wt, A]		32685.1	833470.8	1666941.6	32685.1

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	-0.000	19600.924	-18853.580	97570.958
2	1.0330e-003	-0.000	12291.408	-11744.853	60835.960
3	1.0657e-002	-0.000	7507.616	-7108.727	36864.903
4	9.3936e+000	-0.000	156.866	0.000	113.367
5	2.7248e+001	-0.000	11728.933	7108.727	26251.423
6	1.9304e+001	-0.000	22987.724	13908.379	51449.797
7	1.3095e+001	-0.000	34696.361	21017.105	77781.916
8	2.3136e+001	-0.000	49916.210	30289.358	112112.866
9	7.7924e+000	-0.000	61545.272	37398.085	138415.617
10	1.8943e-002	-0.000	73162.529	44506.812	164724.355
11	5.5000e-005	-0.000	84762.242	51615.538	191042.894

Stresses: (N/mm²)

[σmz, σmb, τmt, τms]	-0.000	0.000	30.964	0.000
[σaz, σab, τat, τas]	0.000	101.698	0.000	7.793
[σzmax, σbmax, τtmax, τsmax]	-0.000	172.886	52.639	13.249

FATIQUE PROOF:
Total safety factor according chapter 2.5.3 [jD] 1.350
(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	2.253	2.086	1.472	1.236
References stress slope	[G']	5.367	5.367	2.300	2.300
Support number	[n(r)]	1.248	1.248	1.265	1.265
Support number	[n(d)]	1.005	1.005	1.007	1.007
Mechanical material support factor	[nwm]	1.059	1.059	1.059	1.059
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	1.805	1.664	1.156	1.000
Roughness factor	[KR]	0.883	0.883	0.932	0.932
Surface stabilization factor	[KV]	1.200	1.200	1.100	1.100
Design coefficient	[KWK]	1.615	1.497	1.117	0.975
Fatigue limit of part (N/mm²)	[SWK]	192.916	208.149	162.801	186.411

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.952	0.956	0.981	0.983
Permissible amplitude (N/mm²)	[SAK]	183.656	198.889	159.714	183.324
Effective Miner sum	[DM]	0.3	0.33	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.038	1.038
Permissible amplitude (N/mm²)	[SBK]	183.656	198.889	165.794	190.302
Rate of utilization	[aBK]	0.000	0.690	0.000	0.055

Calculation of the combined stress types:
Rate of utilization for the combined load components
a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	53.632
Rate of utilization	[aBKv_1]	0.690

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	53.632
Rate of utilization	[aBKv_2]	0.055

Highest utilization	[aBKmax]	0.690
Safety endurance limit assessment	[S.Dauer]	1.956
Required safety	[jD]	1.350
Result (%)	[S/jD]	144.9

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.700	1.330	
Plastic support number	[npl]	1.0000	1.4436	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	551.83	796.62	423.73	318.60
Rate of utilization	[aSK]	0.000	0.304	0.174	0.058

Rate of utilization for the combined load components:
a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	195.454
Rate of utilization	[aSKvBn]	0.350

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	114.121
Rate of utilization	[aSKvQn]	0.232

Highest utilization	[aSKmax]	0.350

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	5.646
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	305.2

Safety against yield point	[S.Rp]	3.999

Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	285.6

Cross section 'B-B' Shoulder

Comment	Y= 300.00mm			
Position (Y-Coordinate) (mm)		[y]		300.000
External diameter (mm)		[da]		187.000
Inner diameter (mm)		[di]		0.000
Notch effect			Shoulder	
[D, r, t] (mm)	205.000	6.000	9.000	
Mean roughness (µm)		[Rz]		8.000

	Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)				
Mean value [Fzdm, Mbm, Tm, Fqm]	-0.0	0.0	-0.0	0.0
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	71039.1	0.0	320967.8
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	-0.0120766.4		0.0	545645.2
Cross section, moment of resistance: (mm²)				
[A, Wb, Wt, A]	27464.6	641984.8	1283969.5	27464.6

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	-0.000	19962.352	-0.000	88519.163
2	1.0330e-003	-0.000	12511.174	-0.000	55208.440
3	1.0657e-002	-0.000	7630.802	0.000	33513.007
4	9.3936e+000	-0.000	133.382	-0.000	363.195
5	2.7248e+001	-0.000	9859.560	-0.000	43886.759
6	1.9304e+001	-0.000	19307.700	0.000	86219.345
7	1.3095e+001	-0.000	29122.582	-0.000	130482.142
8	2.3136e+001	-0.000	41870.801	-0.000	188225.007
9	7.7924e+000	-0.000	51610.677	0.000	232442.029
10	1.8943e-002	-0.000	61333.030	-0.000	276708.455
11	5.5000e-005	-0.000	71039.085	-0.000	320967.762

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	-0.000	0.000	-0.000	0.000
[σaz, σab, τat, τas]	0.000	110.655	0.000	15.582
[σzmax, σbmax, τtmax, τsmax]	-0.000	188.114	0.000	26.490

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	2.224	2.061	1.497	1.248
References stress slope	[G']	0.439	0.439	0.192	0.192
Support number	[n(r)]	1.108	1.108	1.094	1.094
Support number	[n(d)]	1.006	1.006	1.007	1.007
Mechanical material support factor	[nwm]	1.059	1.059	1.059	1.059
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.007	1.850	1.358	1.141
Roughness factor	[KR]	0.883	0.883	0.932	0.932
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.100
Design coefficient	[KWK]	1.784	1.652	1.192	1.103
Fatigue limit of part (N/mm²)	[SWK]	174.713	188.586	152.491	164.788

Calculation with individual mean stress:

Mean stress coefficient	[KAK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	174.713	188.586	152.491	164.788
Effective Miner sum	[DM]	0.3	0.331	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.038
Permissible amplitude (N/mm²)	[SBK]	174.713	188.586	152.491	171.061
Rate of utilization	[aBK]	0.000	0.792	0.000	0.123

Calculation of the combined stress types:

Rate of utilization for the combined load components

a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.792

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	0.000
Rate of utilization	[aBKv_2]	0.123

Highest utilization	[aBKmax]	0.792
Safety endurance limit assessment	[S.Dauer]	1.704
Required safety	[jD]	1.350
Result (%)	[S/jD]	126.2

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]	1.700		1.330	
Plastic support number	[npl]	1.0000	1.4436	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	551.83	796.62	423.73	318.60
Rate of utilization	[aSK]	0.000	0.331	0.000	0.116

Rate of utilization for the combined load components:
a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	188.114
Rate of utilization	[aSKvBn]	0.331

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	45.881
Rate of utilization	[aSKvQn]	0.116

Highest utilization [aSKmax] 0.331

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	5.978
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	323.2

Safety against yield point	[S.Rp]	4.235
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	302.5

Cross section 'C-C' Shoulder

Comment Y= 666.00mm

Position (Y-Coordinate) (mm)		[y]		666.000
External diameter (mm)		[da]		183.000
Inner diameter (mm)		[di]		0.000
Notch effect			Shoulder	
[D, r, t] (mm)	204.000	5.000	10.500	
Mean roughness (µm)		[Rz]		8.000

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value	[Fzdm, Mbm, Tm, Fqm]	0.0	0.0	1164.3	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	44992.5	0.0	324071.4
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	0.0	76487.3	1979.2	550921.4
Cross section, moment of resistance: (mm²)					
[A, Wb, Wt, A]		26302.2	601662.8	1203325.6	26302.2

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	0.000	7057.008	-425.269	50495.485
2	1.0330e-003	0.000	4460.748	-264.921	31678.042
3	1.0657e-002	0.000	2752.232	-160.347	19402.112
4	9.3936e+000	0.000	96.920	0.000	560.188
5	2.7248e+001	0.000	6301.926	160.347	44171.185
6	1.9304e+001	0.000	12311.518	313.723	86941.826
7	1.3095e+001	0.000	18533.139	474.070	131654.263
8	2.3136e+001	0.000	26597.367	683.219	189970.384
9	7.7924e+000	0.000	32744.678	843.566	234658.851
10	1.8943e-002	0.000	38877.600	1003.913	279362.513
11	5.5000e-005	0.000	44992.536	1164.260	324071.439

Stresses: (N/mm²)

[σmz, σmb, τmt, τms]	0.000	0.000	0.968	0.000
[σaz, σab, τat, τas]	0.000	74.780	0.000	16.428
[σzmax, σbmax, τtmax, τsmax]	0.000	127.127	1.645	27.928

FATIQUE PROOF:
Total safety factor according chapter 2.5.3 [jD] 1.350
(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	2.403	2.224	1.580	1.290
References stress slope	[G']	0.519	0.519	0.230	0.230
Support number	[n(r)]	1.117	1.117	1.103	1.103
Support number	[n(d)]	1.006	1.006	1.007	1.007
Mechanical material support factor	[nwm]	1.059	1.059	1.059	1.059
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.151	1.980	1.421	1.169
Roughness factor	[KR]	0.883	0.883	0.932	0.932
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.100
Design coefficient	[KWK]	1.903	1.761	1.245	1.129
Fatigue limit of part (N/mm²)	[SWK]	163.734	176.997	146.037	161.043
Calculation with principal mean stress:					
Mean stress coefficient	[KAK]	0.998	0.998	0.999	0.999
Permissible amplitude (N/mm²)	[SAK]	163.445	176.708	145.941	160.946
Effective Miner sum	[DM]	0.3	0.333	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.038	1.038
Permissible amplitude (N/mm²)	[SBK]	163.445	176.708	151.496	167.073
Rate of utilization	[aBK]	0.000	0.571	0.000	0.133

Calculation of the combined stress types:
Rate of utilization for the combined load components

a)For outer surface (shear stress = 0)		
Equivalent mean stress (N/mm²)	[SmV_1]	1.676
Rate of utilization	[aBKv_1]	0.571
b) For neutral line (Bending stress = 0)		
Equivalent mean stress (N/mm²)	[SmV_2]	1.676
Rate of utilization	[aBKv_2]	0.133
Highest utilization	[aBKmax]	0.571
Safety endurance limit assessment	[S.Dauer]	2.363
Required safety	[jD]	1.350
Result (%)	[S/jD]	175.0

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.700	1.330	
Plastic support number	[npl]	1.0000	1.4436	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	551.83	796.62	423.73	318.60
Rate of utilization	[aSK]	0.000	0.223	0.005	0.123

Rate of utilization for the combined load components:

a)For outer surface (shear stress = 0)		
Equivalent stress (N/mm²)	[SvBn]	127.158
Rate of utilization	[aSKvBn]	0.223
b) For neutral line (Bending stress = 0)		
Equivalent stress (N/mm²)	[SvQn]	51.221
Rate of utilization	[aSKvQn]	0.128
Highest utilization	[aSKmax]	0.223

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	8.844
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	478.1
Safety against yield point	[S.Rp]	6.264
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	447.5

Cross section 'D-D' Shoulder with relief groove

Comment	Y= 123.00mm					
Position (Y-Coordinate) (mm)	[y]	123.000				
External diameter (mm)	[da]	140.000				
Inner diameter (mm)	[di]	0.000				
Notch effect		Shoulder with relief groove				
[D, d, D1, r, t1] (mm)	187.000	139.200	140.000	1.200	Qu[3].Geo.t	
Shape B						
Mean roughness (µm)	[Rz]	8.000				

	Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)				
Mean value [Fzdm, Mbm, Tm, Fqm]	-0.0	0.0	-0.0	0.0
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	14260.9	0.0	320594.4
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	-0.0	24243.5	0.0	545010.5
Cross section, moment of resistance: (mm²)				
[A, Wb, Wt, A]	15218.4	264799.8	529599.5	15218.4

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	-0.000	4270.765	-0.000	88787.198
2	1.0330e-003	-0.000	2715.521	-0.000	55477.224
3	1.0657e-002	-0.000	1675.148	0.000	33783.012
4	9.3936e+000	-0.000	36.056	-0.000	736.540
5	2.7248e+001	-0.000	2124.646	-0.000	43513.428
6	1.9304e+001	-0.000	4079.929	0.000	85846.014
7	1.3095e+001	-0.000	6060.308	-0.000	130108.810
8	2.3136e+001	-0.000	8588.047	-0.000	187851.674
9	7.7924e+000	-0.000	10501.517	0.000	232068.696
10	1.8943e-002	-0.000	12388.713	-0.000	276335.121
11	5.5000e-005	-0.000	14260.877	-0.000	320594.429

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	-0.000	0.000	-0.000	0.000
[σaz, σab, τat, τas]	0.000	53.855	0.000	28.088
[σzmax, σbmax, τtmax, τsmax]	-0.000	91.554	0.000	47.750

FATIQUE PROOF:
Total safety factor according chapter 2.5.3 [jD] 1.350
(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	4.269	3.725	2.398	1.699
References stress slope	[G']	2.013	2.013	0.958	0.958
Support number	[n(r)]	1.194	1.194	1.211	1.211
Support number	[n(d)]	1.007	1.007	1.010	1.010
Mechanical material support factor	[nwm]	1.059	1.059	1.059	1.059
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	3.576	3.098	1.961	1.403
Roughness factor	[KR]	0.883	0.883	0.932	0.932
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.200
Design coefficient	[KWK]	3.091	2.692	1.695	1.230
Fatigue limit of part (N/mm²)	[SWK]	100.825	115.747	107.246	147.812
Calculation with individual mean stress:					
Mean stress coefficient	[KAK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	100.825	115.747	107.246	147.812
Effective Miner sum	[DM]	0.3	0.349	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.038
Permissible amplitude (N/mm²)	[SBK]	100.825	115.747	107.246	153.438
Rate of utilization	[aBK]	0.000	0.628	0.000	0.247

Calculation of the combined stress types:
Rate of utilization for the combined load components
a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.628

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	0.000

Rate of utilization	[aBKv_2]	0.247
Highest utilization	[aBKmax]	0.628
Safety endurance limit assessment	[S.Dauer]	2.149
Required safety	[jD]	1.350
Result (%)	[S/jD]	159.2

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.700	1.330	
Plastic support number	[npl]	1.0000	1.1597	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	551.83	639.95	423.73	318.60
Rate of utilization	[aSK]	0.000	0.200	0.000	0.210

Rate of utilization for the combined load components:

a)For outer surface (shear stress = 0)		
Equivalent stress (N/mm²)	[SvBn]	91.554
Rate of utilization	[aSKvBn]	0.200
b) For neutral line (Bending stress = 0)		
Equivalent stress (N/mm²)	[SvQn]	82.706
Rate of utilization	[aSKvQn]	0.210
Highest utilization	[aSKmax]	0.210

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	9.419
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	509.2
Safety against yield point	[S.Rp]	6.672
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	476.6

Important remarks concerning strength calculation according to FKM-Guideline:

- Calculation with nominal stresses
- Regulation for proof: Utilization <= 1
- Currently the following restrictions still apply::
  Only for axially symmetrical shafts
- Assumption for calculating the notch factor for shearing:
  ßS = 1.0 + (ßT - 1.0) / 2.0 (according to Prof. Haibach)
- Thread: Determination of notch factor as circumferential groove
- Slight interference fit: determination of the notch factor according to fig. 5.3.11 b) with p = 20MPa
- Proven safety: Effective safety according to special formula,
  condition: safety > required safety or result > 100%

End of Report lines: 1023

A.2.4 KISSsoft Report - Intermediate shaft (inferior branch)

File

Name :	Unnamed		
Changed by:	Joana Mêda de Sousa	on: 09.10.2017	at: 22:27:44

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	IMSFINF
Drawing	
Initial position (mm)	0.000
Length (mm)	881.500
Speed (1/min)	373.85
Sense of rotation: counter clockwise	
Material	18CrNiMo7-6
Young's modulus (N/mm²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m³)	7830.000
Coefficient of thermal expansion (10^-6/K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	180.286
Weight of shaft, including additional masses (kg)	180.286
Mass moment of inertia (kg*m²)	0.806
Momentum of mass GD2 (Nm²)	31.623
Weight towards	(0.000, 0.000, -1.000)
Consider deformations due to shearing	
Shear correction coefficient	1.100
Rolling bearing stiffness is calculated from inner bearing geometry	
Tolerance field: Mean value	
Reference temperature (°C)	40.000

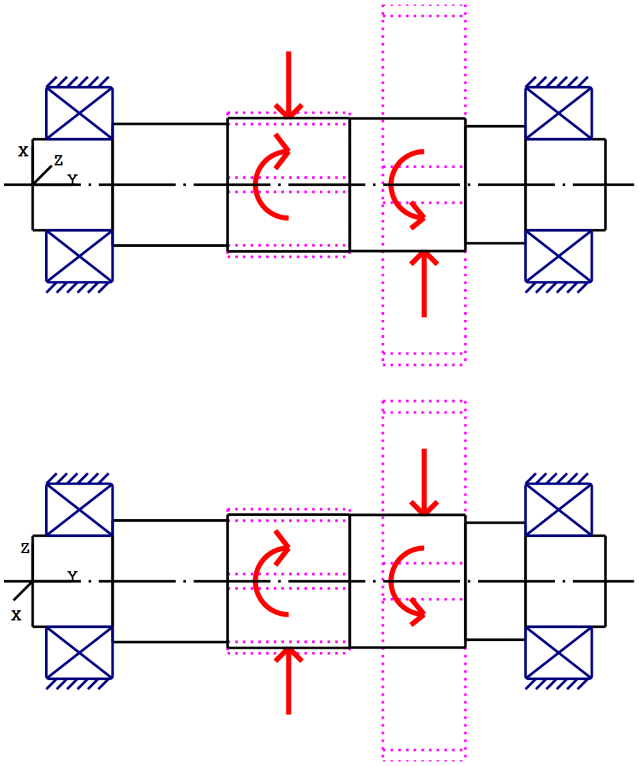


Figure: Load applications

Shaft definition (IMSFINF)

Outer contour

Cylinder (Cylinder)		0.000mm ... 123.000mm
Diameter (mm)	[d]	140.0000
Length (mm)	[l]	123.0000
Surface roughness (µm)	[Rz]	8.0000

Relief groove right (Relief groove right)
r=1.20 (mm), t=0.40 (mm), l=4.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)
Form F (DIN 509), Series 1, with the usual stressing

Chamfer left (Chamfer left)
l=10.00 (mm), alpha=45.00 (°)

Square groove (Square groove)
b=4.00 (mm), t=3.00 (mm), r=0.50 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder)		123.000mm ... 300.000mm
Diameter (mm)	[d]	187.0000
Length (mm)	[l]	177.0000
Surface roughness (µm)	[Rz]	8.0000

Radius right (Radius right)
r=6.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Chamfer left (Chamfer left)

l=10.00 (mm), alpha=45.00 (°)

Cylinder (Cylinder)			300.000mm ...	488.000mm
Diameter (mm)	[d]	205.0000		
Length (mm)	[l]	188.0000		
Surface roughness (µm)	[Rz]	8.0000		

Cylinder (Cylinder)			488.000mm ...	666.000mm
Diameter (mm)	[d]	204.0000		
Length (mm)	[l]	178.0000		
Surface roughness (µm)	[Rz]	8.0000		

Radius left (Radius left)
r=0.50 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Spline (Spline) 439.500mm ... 590.500mm
da=204.00 (mm), df=196.50 (mm), z=67, mn=3.00 (mm), l=151.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Chamfer right (Chamfer right)
l=3.00 (mm), alpha=45.00 (°)

Cylinder (Cylinder)			666.000mm ...	758.500mm
Diameter (mm)	[d]	180.0000		
Length (mm)	[l]	92.5000		
Surface roughness (µm)	[Rz]	8.0000		

Radius left (Radius left)
r=5.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Chamfer right (Chamfer right)
l=8.00 (mm), alpha=45.00 (°)

Cylinder (Cylinder)			758.500mm ...	881.500mm
Diameter (mm)	[d]	140.0000		
Length (mm)	[l]	123.0000		
Surface roughness (µm)	[Rz]	8.0000		

Relief groove left (Relief groove left)
r=1.20 (mm), t=0.40 (mm), l=4.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)
Form F (DIN 509), Series 1, with the usual stressing

Chamfer right (Chamfer right)
l=10.00 (mm), alpha=45.00 (°)

Square groove (Square groove)
b=4.00 (mm), t=3.00 (mm), r=0.50 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Forces

Type of force element		**Cylindrical gear**
Label in the model		IMGINFGEN(IMHSINFPGCT)
Position on shaft (mm)	[y_{local}]	602.5000

Position in global system (mm)	[y_{global}]	602.5000		
Operating pitch diameter (mm)		554.4000		
Helix angle (°)		25.1077 Double helical gearing, right-left		
Working pressure angle at normal section (°)		20.6227		
Position of contact (°)		112.0085		
Length of load application (mm)		127.0000		
Power (kW)		1210.0004		
Torque (Nm)		30907.5080		
Axial force (load spectrum) (N)		0.0000 /	0.0000 /	0.0000
Shearing force X (load spectrum) (N)		-52465.3148 /	-32683.3109 /	-19782.0039
Shearing force Z (Load spectrum) (N)		-51694.7162 /	-32203.2658 /	-19491.4504
Bending moment X (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000
Bending moment Z (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000

Load spectrum, driving (output)

No.	Frequency (%)	Speed (1/min)	Power (kW)	Torque (Nm)
1	1.2440e-003	-373.846	738.100	-18853.580
2	1.0330e-003	-373.846	459.800	-11744.853
3	1.0657e-002	-373.846	278.300	-7108.727
4	9.3936e+000	-373.846	-0.000	0.000
5	2.7248e+001	-373.846	-278.300	7108.727
6	1.9304e+001	-373.846	-544.500	13908.379
7	1.3095e+001	-373.846	-822.800	21017.105
8	2.3136e+001	-373.846	-1185.800	30289.358
9	7.7924e+000	-373.846	-1464.101	37398.085
10	1.8943e-002	-373.846	-1742.401	44506.812
11	5.5000e-005	-373.846	-2020.701	51615.538

Type of force element		**Cylindrical gear**		
Label in the model		IMGINFROT(LSIMINFPGCT)		
Position on shaft (mm)	[y_{local}]	394.0000		
Position in global system (mm)	[y_{global}]	394.0000		
Operating pitch diameter (mm)		221.5516		
Helix angle (°)		25.0871 Double helical gearing, right-left		
Working pressure angle at normal section (°)		20.5050		
Position of contact (°)		30.0000		
Length of load application (mm)		188.0000		
Power (kW)		1210.0004		
Torque (Nm)		-30907.5080		
Axial force (load spectrum) (N)		0.0000 /	0.0000 /	0.0000
Shearing force X (load spectrum) (N)		24233.2773 /	15096.1399 /	9137.1373
Shearing force Z (Load spectrum) (N)		-182534.1178 /	-113709.7783 /	-68824.3395
Bending moment X (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000
Bending moment Z (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000

Load spectrum, driven (input)

No.	Frequency (%)	Speed (1/min)	Power (kW)	Torque (Nm)
1	1.2440e-003	-373.846	-738.100	18853.580
2	1.0330e-003	-373.846	-459.800	11744.853
3	1.0657e-002	-373.846	-278.300	7108.727
4	9.3936e+000	-373.846	0.000	-0.000
5	2.7248e+001	-373.846	278.300	-7108.727
6	1.9304e+001	-373.846	544.500	-13908.379
7	1.3095e+001	-373.846	822.800	-21017.105
8	2.3136e+001	-373.846	1185.800	-30289.358
9	7.7924e+000	-373.846	1464.101	-37398.085
10	1.8943e-002	-373.846	1742.401	-44506.812

11	5.5000e-005	-373.846	2020.701	-51615.538

Bearing

Label in the model		IMSFBRINFGEN
Bearing type		SKF NCF 2328 ECJB
Bearing type		Cylindrical roller bearing (single row)
		SKF Explorer
Bearing position (mm)	[y_{lokal}]	809.500
Bearing position (mm)	[y_{global}]	809.500
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	140.000
External diameter (mm)	[D]	300.000
Width (mm)	[b]	102.000
Corner radius (mm)	[r]	4.000
Number of rolling bodies	[Z]	11
Rolling body reference circle (mm)	[D_{pw}]	212.652
Diameter rolling body (mm)	[D_w]	54.567
Rolling body length (mm)	[L_{we}]	77.914
Diameter, external race (mm)	[d_o]	267.260
Diameter, internal race (mm)	[d_i]	158.044
Calculation with approximate bearings internal geometry (*)		
Bearing clearance		DIN 620:1988 C0 (82.50 µm)
Basic static load rating (kN)	[C_0]	1530.000
Basic dynamic load rating (kN)	[C]	1250.000
Fatigue load rating (kN)	[C_u]	163.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	1249.750
Basic static load rating (kN)	[C_{0theo}]	1529.726

Label in the model		IMSFBRINFROT
Bearing type		SKF NCF 2328 ECJB
Bearing type		Cylindrical roller bearing (single row)
		SKF Explorer
Bearing position (mm)	[y_{lokal}]	72.000
Bearing position (mm)	[y_{global}]	72.000
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	140.000
External diameter (mm)	[D]	300.000
Width (mm)	[b]	102.000
Corner radius (mm)	[r]	4.000
Number of rolling bodies	[Z]	11
Rolling body reference circle (mm)	[D_{pw}]	212.652
Diameter rolling body (mm)	[D_w]	54.567
Rolling body length (mm)	[L_{we}]	77.914
Diameter, external race (mm)	[d_o]	267.260
Diameter, internal race (mm)	[d_i]	158.044
Calculation with approximate bearings internal geometry (*)		
Bearing clearance		DIN 620:1988 C0 (82.50 µm)
Basic static load rating (kN)	[C_0]	1530.000
Basic dynamic load rating (kN)	[C]	1250.000
Fatigue load rating (kN)	[C_u]	163.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	1249.750
Basic static load rating (kN)	[C_{0theo}]	1529.726

Results

Note: the maximum deflection and torsion of the shaft under torque,
the life modification factor aISO, and the bearing's thinnest lubricant film thickness EHL, are predefined for the first load bin.

Shaft

Maximum deflection (µm)	367.650
Position of the maximum (mm)	394.000
Mass center of gravity (mm)	441.933
Total axial load (N)	0.000
Torsion under torque (°)	0.027

Bearing

Probability of failure	[n]	10.00	%
Axial clearance	[u_A]	10.00	µm
Lubricant	Oil: Castrol Optigear Synthetic X 320		

Lubricant with additive, effect on bearing lifetime confirmed in tests.
Oil lubrication, on-line filtration, ISO4406 -/17/14

Lubricant - service temperature	[T_B]	65.00	°C
Limit for factor aISO	[a_{ISOmax}]	50.00	
Oil level	[h_{oil}]	0.00	mm

Oil injection lubrication

Rolling bearing service life according to ISO/TS 16281:2008

Shaft 'IMSFINF' Rolling bearing 'IMSFBRINFGEN'

Position (Y-coordinate)	[y]	809.50	mm
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	31.276	
Nominal bearing service life	[L_{nh}]	641019.50	h
Modified bearing service life	[L_{nmh}]	1000000.00	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	11.36	

Calculation with approximate bearings internal geometry

Operating bearing clearance	[Pd]	125.982	µm
Reference rating service life	[L_{nrh}]	> 1000000	h
Modified reference rating service life	[L_{nrmh}]	> 1000000	h

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	27.241	0.000	117.741	120.851	-485.449	0.000	86.075	493.021
2	16.959	0.000	73.685	75.611	-230.836	0.000	41.148	234.475
3	10.259	0.000	44.948	46.104	-100.458	0.000	18.368	102.124
4	0.000	0.000	0.887	0.887	-0.121	0.000	0.001	0.121
5	4.027	0.000	-17.380	17.840	22.371	0.000	8.532	23.943
6	7.831	0.000	-34.827	35.697	79.274	0.000	28.984	84.407
7	11.785	0.000	-53.069	54.362	164.125	0.000	58.000	174.072
8	16.912	0.000	-76.856	78.695	303.812	0.000	102.788	320.729

9	20.832	0.000	-95.104	97.359	427.369	0.000	142.271	450.428
10	24.742	0.000	-113.349	116.018	563.032	0.000	186.589	593.145
11	28.657	0.000	-131.601	134.685	708.034	0.000	239.311	747.384

	Displacement of bearing				Misalignment of bearing			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	-33.1730	-84.5405	-126.9070	131.1711	0.485	-0.289	-0.012	0.485
2	-27.6181	-84.5405	-110.9767	114.3617	0.307	-0.180	0.002	0.307
3	-22.5526	-84.5405	-99.1003	101.6341	0.189	-0.109	0.006	0.189
4	6.5737	-84.5405	-68.3251	68.6406	0.003	0.000	0.000	0.003
5	-17.2157	-84.5405	84.6959	86.4279	-0.061	0.109	-0.122	0.137
6	-20.0744	-84.5405	94.3807	96.4920	-0.143	0.213	-0.186	0.234
7	-22.7358	-84.5405	102.7219	105.2079	-0.228	0.322	-0.250	0.338
8	-25.9142	-84.5405	112.3252	115.2758	-0.339	0.465	-0.330	0.473
9	-28.2002	-84.5405	119.1205	122.4130	-0.425	0.574	-0.389	0.576
10	-29.4913	-84.5405	125.5823	128.9986	-0.510	0.683	-0.449	0.680
11	-30.1396	-84.5405	131.8211	135.2228	-0.595	0.792	-0.509	0.784

Shaft 'IMSFINF' Rolling bearing 'IMSFBRINFROT'

Position (Y-coordinate)	[y]	72.00	mm
Life modification factor for reliability$[a_1]$		1.000	
Life modification factor	$[a_{ISO}]$	33.551	
Nominal bearing service life	$[L_{nh}]$	137157.25	h
Modified bearing service life	$[L_{nmh}]$	873256.35	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	$[h_{min}]$	0.000	µm
Static safety factor	$[S_0]$	6.34	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	125.982	µm
Reference rating service life	$[L_{nrh}]$	261818.57	h
Modified reference rating service life	$[L_{nrmh}]$	> 1000000	h

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	0.989	0.000	118.240	118.244	499.631	0.000	-24.571	500.234
2	0.627	0.000	73.984	73.987	236.259	0.000	-10.677	236.500
3	0.385	0.000	45.126	45.128	103.475	0.000	-3.914	103.549
4	0.000	0.000	0.881	0.881	0.122	0.000	-0.001	0.122
5	23.235	0.000	-22.900	32.623	-32.981	0.000	-26.976	42.608
6	45.498	0.000	-45.659	64.457	-113.836	0.000	-96.968	149.538
7	68.804	0.000	-69.466	97.772	-228.536	0.000	-197.434	302.008
8	99.234	0.000	-100.524	141.253	-412.333	0.000	-358.346	546.288
9	122.575	0.000	-124.324	174.588	-562.050	0.000	-494.805	748.821
10	145.925	0.000	-148.127	207.932	-726.165	0.000	-642.753	969.767
11	169.271	0.000	-171.924	241.269	-894.921	0.000	-796.552	1198.073

	Displacement of bearing				Misalignment of bearing			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	2.2918	85.0845	-130.2942	130.3144	-0.489	0.000	0.073	0.495
2	2.0435	85.0845	-114.1117	114.1300	-0.307	0.000	0.056	0.312
3	-1.0387	85.0845	-101.3586	101.3639	-0.188	0.000	0.039	0.192
4	6.5841	85.0845	-68.3121	68.6286	-0.003	-0.000	0.000	0.003
5	-66.9742	85.0845	67.4114	95.0255	0.118	-0.000	0.017	0.119
6	-78.2394	85.0845	76.9920	109.7687	0.210	-0.000	0.085	0.227
7	-88.5187	85.0845	85.3292	122.9497	0.305	-0.000	0.156	0.343
8	-99.4033	85.0845	95.9593	138.1637	0.426	-0.000	0.252	0.495

9	-106.2748	85.0845	104.2839	148.8941	0.518	-0.000	0.327	0.612
10	-112.7108	85.0845	112.3351	159.1317	0.609	-0.000	0.400	0.729
11	-118.8059	85.0845	120.2234	169.0222	0.699	-0.000	0.473	0.845

(*) Note about roller bearings with an approximated bearing geometry:
The internal geometry of these bearings has not been input in the database.
The geometry is back-calculated as specified in ISO 281, from C and C0 (details in the manufacturer's catalog).
For this reason, the geometry may be different from the actual geometry.
This can lead to differences in the service life calculation and, more importantly, the roller bearing stiffness.

Damage (%) [Lreq] (175200.000)

Bin no	B1	B2
1	0.00	0.00
2	0.00	0.00
3	0.00	0.00
4	0.00	0.00
5	0.00	0.01
6	0.01	0.07
7	0.02	0.38
8	0.24	4.37
9	0.25	4.23
10	0.00	0.02
11	0.00	0.00
Σ	0.52	9.08

Utilization (%) [Lreq] (175200.000)

B1	B2
48.17	48.69

Note: Utilization = (Lreq/Lh)^(1/k)
Ball bearing: k = 3, roller bearing: k = 10/3

B1: IMSFBRINFGEN
B2: IMSFBRINFROT

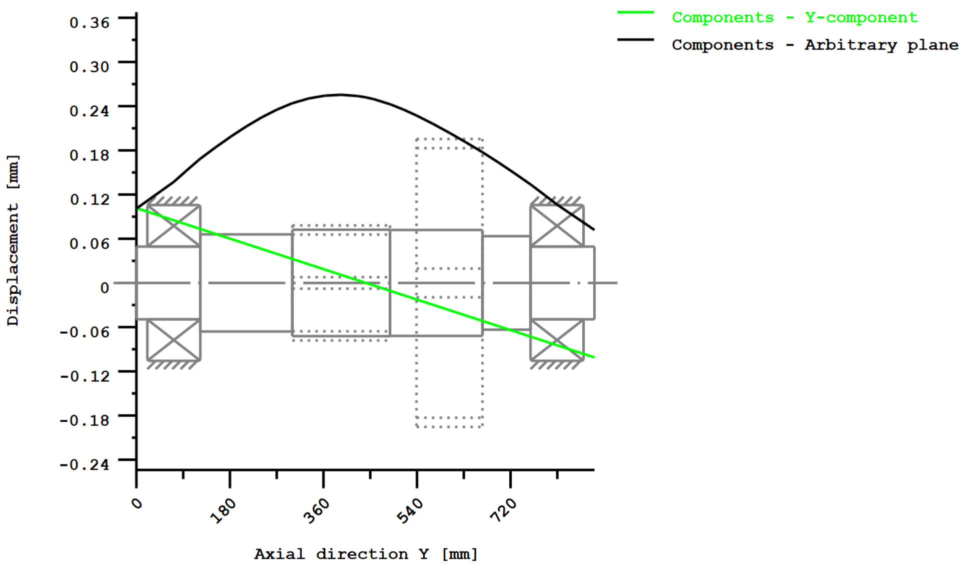
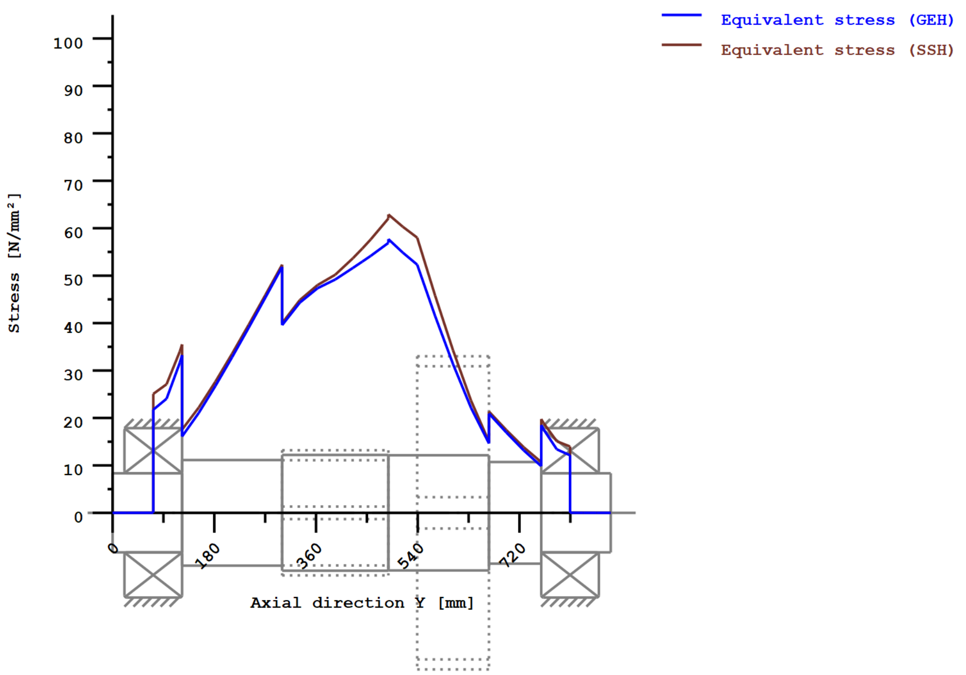


Figure: Deformation (bending etc.) (Arbitrary plane 126.4681308 121)



Nominal stresses, without taking into account stress concentrations
GEH(von Mises): sigV = ((sigB+sigZ,D)^2 + 3*(tauT+tauS)^2)^1/2
SSH(Tresca): sigV = ((sigB-sigZ,D)^2 + 4*(tauT+tauS)^2)^1/2

Figure: Equivalent stress

Strength calculation as specified in the FKM Guideline (6th Edition, 2012)

Summary

IMSFINF

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

Calculation of service strength and static strength

S-N curve (Woehler line) according Miner elementary

Rolled steel, case-hardening steel

Overload case F1 (chapter 2.4.2): Constant mean stress

Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 1.5	[jG]	1.00

Cross section	Pos (Y co-ord) (mm)	
A-A	300.00	Shoulder
B-B	123.00	Shoulder with relief groove
C-C	488.00	Shoulder
D-D	119.00	Interference fit

Results:

Cross section	Kfb	KRs	ALGmax	SD	SS	SB
A-A	1.85	0.88	0.60	2.25	5.59	7.89
B-B	3.10	0.88	0.49	2.76	8.86	12.51
C-C	1.66	0.88	0.43	3.14	5.45	7.69
D-D	2.61	1.00	0.35	3.81	8.96	12.66
Nominal safety:				1.35	1.40	1.85

Abbreviations:

Kfb: Notch factor bending

KRs: Surface factor

ALGmax: Highest utilization

SD: Safety endurance limit

SS: Safety against yield point

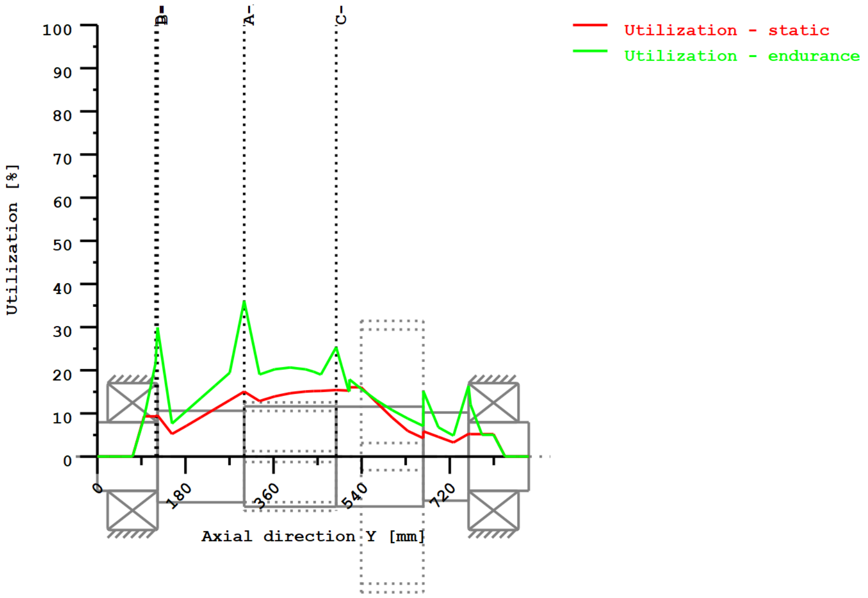
SB: Safety against tensile stress

Service life and damage

System service life (h)	[Hatt]	1000000.00
Damage to system (%)	[D]	0.00

Utilization (%) [Smin/S]

Cross section	Static	Endurance
A-A	25.065	60.058
B-B	15.798	48.953
C-C	25.705	43.002
D-D	15.617	35.450
Maximum utilization (%)	[A]	60.058



Utilization = Smin/S (%)

Figure: Strength

Calculation details

General statements

Label	IMSFINF		
Drawing			
Length (mm)		[l]	881.50
Speed (1/min)		[n]	373.85

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

	Tension/Compression	Bending	Torsion	Shearing
Load factor static calculation	1.700	1.700	1.700	1.700
Load factor endurance limit	1.000	1.000	1.000	1.000

Rolled steel, case-hardening steel

Base stress according FKM chapter 5.1:

Tensile strength (N/mm²)	[Rm,N]	1200.00
Yield point (N/mm²)	[Rp,N]	850.00
Fatigue limit (N/mm²)	[σzdWN]	480.00
Fatigue limit (N/mm²)	[σbWN]	510.00

Fatigue limit (N/mm²)	[τtWN]	305.00		
Fatigue limit (N/mm²)	[τsWN]	280.00		
Breaking elongation (%)	[A]	8.00		
Reference diameter (mm)	[deffNm, deffNp]	16.00	16.00	
Required life time	[H]	175200.00		
Number of load cycles	[NL]	3929870769		
Service strength for a load spectrum				
S-N curve (Woehler lines) according to Miner elementary according to FKM guideline				
Temperature (°C)	[Temperatur]	20.000		
Temperature duration (h)	[TemperaturD]	175200.000		
Temperature coefficients	[KTm, KTp, KTD]	1.000	1.000	1.000
	[KTtm, KTtp]	1.000	1.000	
Internal stress coefficient	[KEs, KEt]	1.000	1.000	
Additional coefficients	[KA, KW, KfW]	1.000	1.000	1.000
	[KNL, KNLE]	1.000	1.000	
Protective layer factor	[KS]	1.000		

Material properties:

[fσZ, fσD, fτ, Rpmax]	1.000	1.000	0.577	1150.0
[fWt, fWs]	0.577	0.400		
[aM, bM, aTD]	0.35000	-0.100	1.400	
[aG, bG, aRsig, RmNmin]	0.500	2700.0	0.220	400.0
[MS, MT]	0.1727	0.0997		
[kσ, kτ]	15	25		
[kDσ, kDτ]	0	0		
[NDσ, NDτ]	1e+006	1e+006		
[NDσII, NDτII]	0	0		

Thickness of raw material (mm)	[d.eff]	210.00	
Material data calculated acc. FKM directive with Kdm, Kdp			
Geometric size factors (Kdm, Kdp) calculated from raw diameter			
Material strength calculated from size of raw material			
Constants	[adm, adp]	0.370	0.370
Size factors	[Kdm, Kdp]	0.649	0.649
Tensile strength (N/mm²)	[Rm]	779.05	
Yield point (N/mm²)	[Rp]	551.83	
σzdW (N/mm²)	[σzdW]	311.62	
σbW (N/mm²)	[σbW]	331.10	
τtW (N/mm²)	[τtW]	198.01	
τsW (N/mm²)	[τsW]	181.78	

Overload case F1 (chapter 2.4.2): Constant mean stress		
Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jG]	1.00

Cross section 'A-A' Shoulder

Comment	Y= 300.00mm		
Position (Y-Coordinate) (mm)		[y]	300.000

External diameter (mm)	[da]	187.000	
Inner diameter (mm)	[di]	0.000	
Notch effect		Shoulder	
[D, r, t] (mm)	205.000	6.000	9.000
Mean roughness (µm)	[Rz]	8.000	

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value	[Fzdm, Mbm, Tm, Fqm]	-0.0	0.0	-0.0	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	53860.5	0.0	241639.1
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	-0.0	91562.9	0.0	410786.5
Cross section, moment of resistance: (mm²)					
[A, Wb, Wt, A]		27464.6	641984.8	1283969.5	27464.6

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	-0.000	26389.858	0.000	117741.437
2	1.0330e-003	-0.000	16563.263	0.000	73480.342
3	1.0657e-002	-0.000	10116.661	0.000	44618.783
4	9.3936e+000	-0.000	133.094	-0.000	362.584
5	2.7248e+001	-0.000	7442.371	-0.000	32996.126
6	1.9304e+001	-0.000	14594.989	-0.000	64825.949
7	1.3095e+001	-0.000	22038.578	-0.000	98141.449
8	2.3136e+001	-0.000	31708.296	-0.000	141622.601
9	7.7924e+000	-0.000	39106.264	-0.000	174958.011
10	1.8943e-002	-0.000	46487.881	-0.000	208302.022
11	5.5000e-005	-0.000	53860.506	-0.000	241639.099

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	-0.000	0.000	-0.000	0.000
[σaz, σab, τat, τas]	0.000	83.897	0.000	11.731
[σzmax, σbmax, τtmax, τsmax]	-0.000	142.625	0.000	19.943

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	2.224	2.061	1.497	1.248
References stress slope	[G']	0.439	0.439	0.192	0.192
Support number	[n(r)]	1.108	1.108	1.094	1.094
Support number	[n(d)]	1.006	1.006	1.007	1.007
Mechanical material support factor	[nwm]	1.059	1.059	1.059	1.059
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.007	1.850	1.358	1.141
Roughness factor	[KR]	0.883	0.883	0.932	0.932
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.100
Design coefficient	[KWK]	1.784	1.652	1.192	1.103
Fatigue limit of part (N/mm²)	[SWK]	174.713	188.586	152.491	164.788

Calculation with individual mean stress:

Mean stress coefficient	[KAK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	174.713	188.586	152.491	164.788

Effective Miner sum	[DM]	0.3	0.33	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.026	1.038
Permissible amplitude (N/mm²)	[SBK]	174.713	188.586	156.484	171.061
Rate of utilization	[aBK]	0.000	0.601	0.000	0.093

Calculation of the combined stress types:
Rate of utilization for the combined load components
a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.601

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	0.000
Rate of utilization	[aBKv_2]	0.093

Highest utilization	[aBKmax]	0.601
Safety endurance limit assessment	[S.Dauer]	2.248
Required safety	[jD]	1.350
Result (%)	[S/jD]	166.5

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.700	1.330	
Plastic support number	[npl]	1.0000	1.4436	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	551.83	796.62	423.73	318.60
Rate of utilization	[aSK]	0.000	0.251	0.000	0.088

Rate of utilization for the combined load components:
a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	142.625
Rate of utilization	[aSKvBn]	0.251

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	34.542
Rate of utilization	[aSKvQn]	0.088

Highest utilization	[aSKmax]	0.251

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	7.885
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	426.2
Safety against yield point	[S.Rp]	5.585
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	399.0

Cross section 'B-B' Shoulder with relief groove

Comment	Y= 123.00mm	
Position (Y-Coordinate) (mm)	[y]	123.000
External diameter (mm)	[da]	140.000
Inner diameter (mm)	[di]	0.000
Notch effect		Shoulder with relief groove

[D, d, D1, r, t1] (mm)	187.000	139.200	140.000	1.200	Qu[1].Geo.t

Shape B

Mean roughness (µm) [Rz] 8.000

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value	[Fzdm, Mbm, Tm, Fqm]	-0.0	0.0	-0.0	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	11114.1	0.0	241372.8
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	-0.0	18894.0	0.0	410333.8
Cross section, moment of resistance: (mm²)					
	[A, Wb, Wt, A]	15218.4	264799.8	529599.5	15218.4

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	-0.000	5516.615	0.000	118114.769
2	1.0330e-003	-0.000	3524.212	0.000	73853.673
3	1.0657e-002	-0.000	2186.099	0.000	44992.114
4	9.3936e+000	-0.000	35.876	-0.000	735.930
5	2.7248e+001	-0.000	1625.494	-0.000	32732.136
6	1.9304e+001	-0.000	3144.344	-0.000	64560.534
7	1.3095e+001	-0.000	4691.142	-0.000	97875.567
8	2.3136e+001	-0.000	6664.750	-0.000	141356.476
9	7.7924e+000	-0.000	8162.371	-0.000	174691.803
10	1.8943e-002	-0.000	9642.124	-0.000	208035.764
11	5.5000e-005	-0.000	11114.109	-0.000	241372.807

Stresses: (N/mm²)

[σmz, σmb, τmt, τms]	-0.000	0.000	-0.000	0.000
[σaz, σab, τat, τas]	0.000	41.972	0.000	21.147
[σzmax, σbmax, τtmax, τsmax]	-0.000	71.352	0.000	35.951

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	4.269	3.725	2.398	1.699
References stress slope	[G']	2.013	2.013	0.958	0.958
Support number	[n(r)]	1.194	1.194	1.211	1.211
Support number	[n(d)]	1.007	1.007	1.010	1.010
Mechanical material support factor	[nwm]	1.059	1.059	1.059	1.059
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	3.576	3.098	1.961	1.403
Roughness factor	[KR]	0.883	0.883	0.932	0.932
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.200
Design coefficient	[KWK]	3.091	2.692	1.695	1.230
Fatigue limit of part (N/mm²)	[SWK]	100.825	115.747	107.246	147.812

Calculation with individual mean stress:

Mean stress coefficient	[KAK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	100.825	115.747	107.246	147.812
Effective Miner sum	[DM]	0.3	0.345	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.026	1.038

Permissible amplitude (N/mm²)	[SBK]	100.825	115.747	110.055	153.438
Rate of utilization	[aBK]	0.000	0.490	0.000	0.186

Calculation of the combined stress types:
Rate of utilization for the combined load components

a)For outer surface (shear stress = 0)		
Equivalent mean stress (N/mm²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.490
b) For neutral line (Bending stress = 0)		
Equivalent mean stress (N/mm²)	[SmV_2]	0.000
Rate of utilization	[aBKv_2]	0.186
Highest utilization	[aBKmax]	0.490
Safety endurance limit assessment	[S.Dauer]	2.758
Required safety	[jD]	1.350
Result (%)	[S/jD]	204.3

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.700	1.330	
Plastic support number	[npl]	1.0000	1.1597	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	551.83	639.95	423.73	318.60
Rate of utilization	[aSK]	0.000	0.156	0.000	0.158

Rate of utilization for the combined load components:

a)For outer surface (shear stress = 0)		
Equivalent stress (N/mm²)	[SvBn]	71.352
Rate of utilization	[aSKvBn]	0.156
b) For neutral line (Bending stress = 0)		
Equivalent stress (N/mm²)	[SvQn]	62.268
Rate of utilization	[aSKvQn]	0.158
Highest utilization	[aSKmax]	0.158

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	12.511
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	676.3
Safety against yield point	[S.Rp]	8.862
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	633.0

Cross section 'C-C' Shoulder

Comment	Y= 488.00mm			
Position (Y-Coordinate) (mm)		[y]		488.000
External diameter (mm)		[da]		204.000
Inner diameter (mm)		[di]		0.000
Notch effect			Shoulder	
[D, r, t] (mm)	205.000	0.500	0.500	
Mean roughness (µm)		[Rz]		8.000

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value	[Fzdm, Mbm, Tm, Fqm]	-0.0	0.0	51615.5	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	52803.1	0.0	266611.2
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	-0.0	89765.2	87746.4	453239.0
Cross section, moment of resistance: (mm²)					
	[A, Wb, Wt, A]	32685.1	833470.8	1666941.6	32685.1

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	-0.000	31433.852	-18853.580	69977.597
2	1.0330e-003	-0.000	19713.800	-11744.853	43639.909
3	1.0657e-002	-0.000	12032.929	-7108.727	26456.975
4	9.3936e+000	-0.000	156.463	0.000	113.978
5	2.7248e+001	-0.000	7260.610	7108.727	36748.620
6	1.9304e+001	-0.000	14272.656	13908.379	71900.768
7	1.3095e+001	-0.000	21574.544	21017.105	108631.303
8	2.3136e+001	-0.000	31064.046	30289.358	156511.103
9	7.7924e+000	-0.000	38323.842	37398.085	193215.218
10	1.8943e-002	-0.000	45568.561	44506.812	229910.073
11	5.5000e-005	-0.000	52803.081	51615.538	266611.198

Stresses: (N/mm²)

[σmz, σmb, τmt, τms]	-0.000	0.000	30.964	0.000
[σaz, σab, τat, τas]	0.000	63.353	0.000	10.876
[σzmax, σbmax, τtmax, τsmax]	-0.000	107.701	52.639	18.489

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	2.253	2.086	1.472	1.236
References stress slope	[G']	5.367	5.367	2.300	2.300
Support number	[n(r)]	1.248	1.248	1.265	1.265
Support number	[n(d)]	1.005	1.005	1.007	1.007
Mechanical material support factor	[nwm]	1.059	1.059	1.059	1.059
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	1.805	1.664	1.156	1.000
Roughness factor	[KR]	0.883	0.883	0.932	0.932
Surface stabilization factor	[KV]	1.200	1.200	1.100	1.100
Design coefficient	[KWK]	1.615	1.497	1.117	0.975
Fatigue limit of part (N/mm²)	[SWK]	192.916	208.149	162.801	186.411

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.952	0.956	0.981	0.983
Permissible amplitude (N/mm²)	[SAK]	183.656	198.889	159.714	183.324
Effective Miner sum	[DM]	0.3	0.33	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.038	1.038
Permissible amplitude (N/mm²)	[SBK]	183.656	198.889	165.794	190.302
Rate of utilization	[aBK]	0.000	0.430	0.000	0.077

Calculation of the combined stress types:
Rate of utilization for the combined load components

a)For outer surface (shear stress = 0)		
Equivalent mean stress (N/mm²)	[SmV_1]	53.632
Rate of utilization	[aBKv_1]	0.430
b) For neutral line (Bending stress = 0)		
Equivalent mean stress (N/mm²)	[SmV_2]	53.632
Rate of utilization	[aBKv_2]	0.077
Highest utilization	[aBKmax]	0.430
Safety endurance limit assessment	[S.Dauer]	3.139
Required safety	[jD]	1.350
Result (%)	[S/jD]	232.5

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.700	1.330	
Plastic support number	[npl]	1.0000	1.4436	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	551.83	796.62	423.73	318.60
Rate of utilization	[aSK]	0.000	0.189	0.174	0.081

Rate of utilization for the combined load components:

a)For outer surface (shear stress = 0)		
Equivalent stress (N/mm²)	[SvBn]	141.110
Rate of utilization	[aSKvBn]	0.257
b) For neutral line (Bending stress = 0)		
Equivalent stress (N/mm²)	[SvQn]	123.198
Rate of utilization	[aSKvQn]	0.255
Highest utilization	[aSKmax]	0.257

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	7.689
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	415.6
Safety against yield point	[S.Rp]	5.446
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	389.0

Cross section 'D-D' Interference fit

Comment Y= 25.00...119.00mm

Position (Y-Coordinate) (mm)	[y]	119.000
External diameter (mm)	[da]	140.000
Inner diameter (mm)	[di]	0.000
Notch effect		Interference fit
Characteristics:	Slight interference fit	
Mean roughness (µm)	[Rz]	8.000

Tension/Compression Bending Torsion Shearing

Load: (N) (Nm)

Mean value [Fzdm, Mbm, Tm, Fqm]		-0.0	0.0	-0.0	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	10148.6	0.0	241369.4
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	-0.0	17252.7	0.0	410328.0

Cross section, moment of resistance: (mm²)

[A, Wb, Wt, A]	15393.8	269391.6	538783.1	15393.8

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	-0.000	5044.151	0.000	118119.498
2	1.0330e-003	-0.000	3228.789	0.000	73858.402
3	1.0657e-002	-0.000	2006.121	0.000	44996.843
4	9.3936e+000	-0.000	32.922	-0.000	740.659
5	2.7248e+001	-0.000	1494.572	-0.000	32728.806
6	1.9304e+001	-0.000	2886.110	-0.000	64557.179
7	1.3095e+001	-0.000	4299.650	-0.000	97872.204
8	2.3136e+001	-0.000	6099.339	-0.000	141353.108
9	7.7924e+000	-0.000	7463.620	-0.000	174688.434
10	1.8943e-002	-0.000	8810.000	-0.000	208032.393
11	5.5000e-005	-0.000	10148.639	-0.000	241369.436

Stresses: (N/mm²)

[σmz, σmb, τmt, τms]	-0.000	0.000	-0.000	0.000
[σaz, σab, τat, τas]	0.000	37.672	0.000	20.906
[σzmax, σbmax, τtmax, τsmax]	-0.000	64.043	0.000	35.541

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Notch effect coefficient	[β(dB)]	2.458	2.458	1.656	1.328
[dB] (mm) 40.0, [rB] (mm) 2.4, [r] (mm) 8.4					
Support number	[n(r)]	1.097	1.097	1.074	1.074
Support number	[n(rB)]	1.172	1.172	1.139	1.139
Support number	[n(d)]	1.007	1.007	1.010	1.010
Mechanical material support factor	[nwm]	1.059	1.059	1.059	1.059
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.626	2.607	1.739	1.408
Roughness factor	[KR]	1.000	1.000	1.000	1.000
Roughness factor is included into the notch effect coefficient					
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.200
Design coefficient	[KWK]	2.188	2.172	1.449	1.173
Fatigue limit of part (N/mm²)	[SWK]	142.416	143.463	125.435	154.912

Calculation with individual mean stress:

Mean stress coefficient	[KAK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	142.416	143.463	125.435	154.912
Effective Miner sum	[DM]	0.3	0.347	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.026	1.038
Permissible amplitude (N/mm²)	[SBK]	142.416	143.463	128.720	160.809
Rate of utilization	[aBK]	0.000	0.355	0.000	0.176

Calculation of the combined stress types:

Rate of utilization for the combined load components

a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.355

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	0.000
Rate of utilization	[aBKv_2]	0.176

Highest utilization	[aBKmax]	0.355
Safety endurance limit assessment	[S.Dauer]	3.808
Required safety	[jD]	1.350
Result (%)	[S/jD]	282.1

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.400

(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.700	1.330	
Plastic support number	[npl]	1.0000	1.4436	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	551.83	796.62	423.73	318.60
Rate of utilization	[aSK]	0.000	0.113	0.000	0.156

Rate of utilization for the combined load components:

a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	64.043
Rate of utilization	[aSKvBn]	0.113

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	61.558
Rate of utilization	[aSKvQn]	0.156

Highest utilization	[aSKmax]	0.156

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	12.655
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	684.1

Safety against yield point	[S.Rp]	8.964
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	640.3

Important remarks concerning strength calculation according to FKM-Guideline:

- Calculation with nominal stresses
- Regulation for proof: Utilization <= 1
- Currently the following restrictions still apply::
  Only for axially symmetrical shafts
- Assumption for calculating the notch factor for shearing:
  ßS = 1.0 + (ßT - 1.0) / 2.0 (according to Prof. Haibach)
- Thread: Determination of notch factor as circumferential groove
- Slight interference fit: determination of the notch factor according to fig. 5.3.11 b) with p = 20MPa
- Proven safety: Effective safety according to special formula,
  condition: safety > required safety or result > 100%

End of Report	lines:	1009

A.2.5 KISSsoft Report - High speed shaft (Output)

File

Name : Unnamed
Changed by: Joana Mêda de Sousa on: 09.10.2017 at: 22:55:33

Important hint: At least one warning has occurred during the calculation:

1-> Cross section B-B:
Please note: The static maximum rate of utilization is greater than the dynamic rate, so the static strength verification is obligatory!

Analysis of shafts, axle and beams

Input data

Coordinate system shaft: see picture W-002

Label	HSSF
Drawing	
Initial position (mm)	0.000
Length (mm)	1081.000
Speed (1/min)	1495.38
Sense of rotation: clockwise	
Material	18CrNiMo7-6
Young's modulus (N/mm²)	206000.000
Poisson's ratio nu	0.300
Density (kg/m³)	7830.000
Coefficient of thermal expansion (10^-6/K)	11.500
Temperature (°C)	20.000
Weight of shaft (kg)	88.576
Weight of shaft, including additional masses (kg)	88.576
Mass moment of inertia (kg*m²)	0.158
Momentum of mass GD2 (Nm²)	6.187
Weight towards	(0.000, 0.000, -1.000)
Consider deformations due to shearing	
Shear correction coefficient	1.100
Rolling bearing stiffness is calculated from inner bearing geometry	
Tolerance field: Mean value	
Reference temperature (°C)	40.000

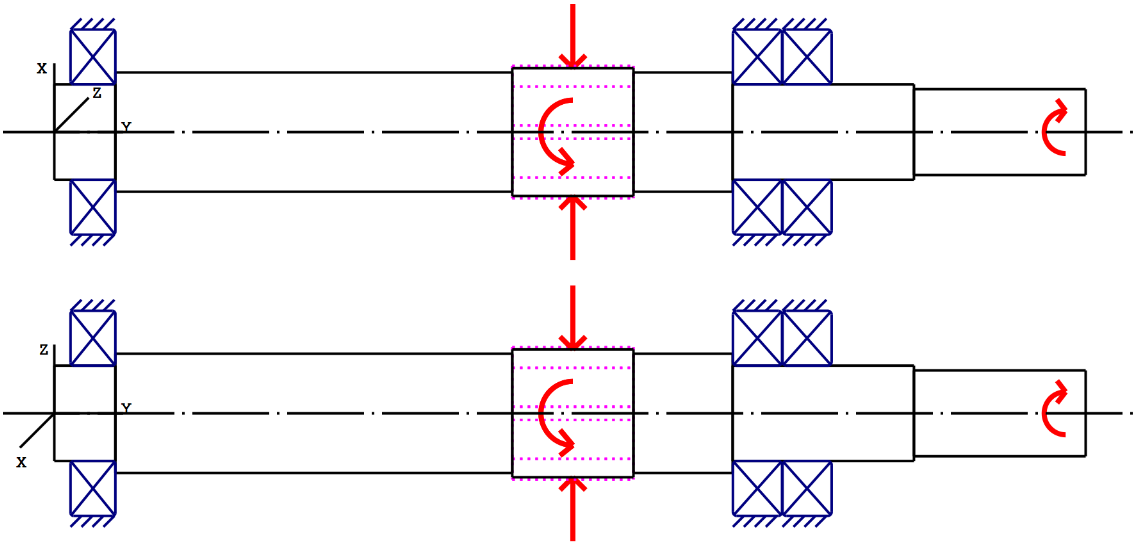


Figure: Load applications

Shaft definition (HSSF)

Outer contour

Cylinder (Cylinder)			0.000mm ... 64.000mm
Diameter (mm)	[d]	100.0000	
Length (mm)	[l]	64.0000	
Surface roughness (µm)	[Rz]	8.0000	

Chamfer left (Chamfer left)
l=8.00 (mm), alpha=45.00 (°)

Square groove (Square groove)
b=3.00 (mm), t=2.00 (mm), r=0.50 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Relief groove right (Relief groove right)
r=1.20 (mm), t=0.40 (mm), l=4.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)
Form F (DIN 509), Series 1, with the usual stressing

Cylinder (Cylinder)			64.000mm ... 480.000mm
Diameter (mm)	[d]	125.0000	
Length (mm)	[l]	416.0000	
Surface roughness (µm)	[Rz]	8.0000	

Chamfer left (Chamfer left)
l=5.00 (mm), alpha=45.00 (°)

Radius right (Radius right)

r=1.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Cylinder (Cylinder)			480.000mm ... 607.000mm
Diameter (mm)	[d]	134.0000	
Length (mm)	[l]	127.0000	
Surface roughness (µm)	[Rz]	8.0000	

Cylinder (Cylinder)			607.000mm ... 711.000mm
Diameter (mm)	[d]	125.0000	
Length (mm)	[l]	104.0000	
Surface roughness (µm)	[Rz]	8.0000	

Radius left (Radius left)
r=1.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Chamfer right (Chamfer right)
l=5.00 (mm), alpha=45.00 (°)

Cylinder (Cylinder)			711.000mm ... 901.000mm
Diameter (mm)	[d]	100.0000	
Length (mm)	[l]	190.0000	
Surface roughness (µm)	[Rz]	8.0000	

Relief groove left (Relief groove left)
r=1.20 (mm), t=0.40 (mm), l=4.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)
Form F (DIN 509), Series 1, with the usual stressing

Chamfer right (Chamfer right)
l=2.00 (mm), alpha=45.00 (°)

Cylinder (Cylinder)			901.000mm ... 1081.000mm
Diameter (mm)	[d]	90.0000	
Length (mm)	[l]	180.0000	
Surface roughness (µm)	[Rz]	8.0000	

Key way (Key way) 936.000mm ... 1046.000mm
l=110.00 (mm), Rz=8.0, Turned (Ra=3.2µm/125µin)

Forces

Type of force element		**Coupling**		
Label in the model		HSC(HSOUTPUT)		
Position on shaft (mm)	[y_{local}]	1060.0000		
Position in global system (mm)	[y_{global}]	1060.0000		
Effective diameter (mm)		90.0000		
Radial force factor (-)		0.0000		
Direction of the radial force (°)		0.0000		
Axial force factor (-)		0.0000		
Length of load application (mm)		10.0000		
Power (kW)		2420.0081		
Torque (Nm)		-15453.8000		
Axial force (load spectrum) (N)		0.0000 /	0.0000 /	0.0000

Shearing force X (load spectrum) (N)	0.0000 /	0.0000 /	0.0000
Shearing force Z (Load spectrum) (N)	0.0000 /	0.0000 /	0.0000
Mass (kg)	0.0000		
Mass moment of inertia Jp (kg*m²)	0.0000		
Mass moment of inertia Jxx (kg*m²)	0.0000		
Mass moment of inertia Jzz (kg*m²)	0.0000		
Eccentricity (mm)	0.0000		

Load spectrum, driving (output)

No.	Frequency (%)	Speed (1/min)	Power (kW)	Torque (Nm)
1	1.2440e-003	1495.385	1476.205	9426.818
2	1.0330e-003	1495.385	919.603	5872.444
3	1.0657e-002	1495.385	556.602	3554.374
4	9.3936e+000	1495.385	-0.000	-0.000
5	2.7248e+001	1495.385	-556.602	-3554.374
6	1.9304e+001	1495.385	-1089.004	-6954.210
7	1.3095e+001	1495.385	-1645.605	-10508.584
8	2.3136e+001	1495.385	-2371.608	-15144.724
9	7.7924e+000	1495.385	-2928.210	-18699.098
10	1.8943e-002	1495.385	-3484.812	-22253.472
11	5.5000e-005	1495.385	-4041.413	-25807.846

Type of force element		**Cylindrical gear**		
Label in the model		HSG(IMHSINFPGCT)		
Position on shaft (mm)	[y_{local}]	543.5000		
Position in global system (mm)	[y_{global}]	543.5000		
Operating pitch diameter (mm)		138.6000		
Helix angle (°)		25.1077 Double helical gearing, left-right		
Working pressure angle at normal section (°)		20.6227		
Position of contact (°)		-67.9915		
Length of load application (mm)		127.0000		
Power (kW)		1210.0004		
Torque (Nm)		7726.8770		
Axial force (load spectrum) (N)		0.0000 /	0.0000 /	0.0000
Shearing force X (load spectrum) (N)		52465.3148 /	32683.3108 /	19782.0039
Shearing force Z (Load spectrum) (N)		51694.7163 /	32203.2659 /	19491.4504
Bending moment X (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000
Bending moment Z (Load spectrum) (Nm)		0.0000 /	0.0000 /	0.0000

Load spectrum, driven (input)

No.	Frequency (%)	Speed (1/min)	Power (kW)	Torque (Nm)
1	1.2440e-003	1495.385	-738.100	-4713.395
2	1.0330e-003	1495.385	-459.800	-2936.213
3	1.0657e-002	1495.385	-278.300	-1777.182
4	9.3936e+000	1495.385	0.000	0.000
5	2.7248e+001	1495.385	278.300	1777.182
6	1.9304e+001	1495.385	544.500	3477.095
7	1.3095e+001	1495.385	822.800	5254.276
8	2.3136e+001	1495.385	1185.800	7572.339
9	7.7924e+000	1495.385	1464.101	9349.521
10	1.8943e-002	1495.385	1742.401	11126.703
11	5.5000e-005	1495.385	2020.701	12903.885

Type of force element		**Cylindrical gear**
Label in the model		HSG(IMHSSUPPGCT)
Position on shaft (mm)	[y_{local}]	543.5000
Position in global system (mm)	[y_{global}]	543.5000

Operating pitch diameter (mm)	138.6000		
Helix angle (°)	25.1077 Double helical gearing, left-right		
Working pressure angle at normal section (°)	20.6227		
Position of contact (°)	67.9915		
Length of load application (mm)	127.0000		
Power (kW)	1210.0004		
Torque (Nm)	7726.8770		
Axial force (load spectrum) (N)	0.0000 /	0.0000 /	0.0000
Shearing force X (load spectrum) (N)	-73650.7723 /	-45880.8090 /	-27769.9633
Shearing force Z (Load spectrum) (N)	-718.8058 /	-447.7807 /	-271.0251
Bending moment X (Load spectrum) (Nm)	0.0000 /	0.0000 /	0.0000
Bending moment Z (Load spectrum) (Nm)	0.0000 /	0.0000 /	0.0000

Load spectrum, driven (input)

No.	Frequency (%)	Speed (1/min)	Power (kW)	Torque (Nm)
1	1.2440e-003	1495.385	-738.100	-4713.395
2	1.0330e-003	1495.385	-459.800	-2936.213
3	1.0657e-002	1495.385	-278.300	-1777.182
4	9.3936e+000	1495.385	0.000	0.000
5	2.7248e+001	1495.385	278.300	1777.182
6	1.9304e+001	1495.385	544.500	3477.095
7	1.3095e+001	1495.385	822.800	5254.276
8	2.3136e+001	1495.385	1185.800	7572.339
9	7.7924e+000	1495.385	1464.101	9349.521
10	1.8943e-002	1495.385	1742.401	11126.703
11	5.5000e-005	1495.385	2020.701	12903.885

Bearing

Label in the model	HSSFBRGEN1
Bearing type	SKF 30320 J2
Bearing type	Taper roller bearing (single row)

Bearing position (mm)	[y_{lokal}]	737.000
Bearing position (mm)	[y_{global}]	737.000
Attachment of external ring		Set fixed bearing left
Inner diameter (mm)	[d]	100.000
External diameter (mm)	[D]	215.000
Width (mm)	[b]	51.500
Corner radius (mm)	[r]	4.000
Number of rolling bodies	[Z]	14
Rolling body reference circle (mm)	[D_{pw}]	158.020
Diameter rolling body (mm)	[D_w]	28.875
Rolling body length (mm)	[L_{we}]	34.427
Distance a (mm)	[a]	40.161
Diameter, external race (mm)	[d_o]	186.344
Diameter, internal race (mm)	[d_i]	129.696

Calculation with approximate bearings internal geometry (*)

Bearing clearance 0.00 µm

The bearing pressure angle will be considered in the calculation

Position (center of pressure) (mm)

751.4110

Basic static load rating (kN)	[C_0]	490.000
Basic dynamic load rating (kN)	[C]	402.000
Fatigue load rating (kN)	[C_u]	53.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	402.038

Basic static load rating (kN)	[C_{0theo}]	490.045

Label in the model		HSSFBRGEN2
Bearing type		SKF 30320 J2
Bearing type		Taper roller bearing (single row)
Bearing position (mm)	[y_{lokal}]	789.000
Bearing position (mm)	[y_{global}]	789.000
Attachment of external ring		Set fixed bearing right
Inner diameter (mm)	[d]	100.000
External diameter (mm)	[D]	215.000
Width (mm)	[b]	51.500
Corner radius (mm)	[r]	4.000
Number of rolling bodies	[Z]	14
Rolling body reference circle (mm)	[D_{pw}]	158.020
Diameter rolling body (mm)	[D_w]	28.875
Rolling body length (mm)	[L_{we}]	34.427
Distance a (mm)	[a]	40.161
Diameter, external race (mm)	[d_o]	186.344
Diameter, internal race (mm)	[d_i]	129.696

Calculation with approximate bearings internal geometry (*)

Bearing clearance 0.00 µm

The bearing pressure angle will be considered in the calculation

Position (center of pressure) (mm)

774.5890

Basic static load rating (kN)	[C_0]	490.000
Basic dynamic load rating (kN)	[C]	402.000
Fatigue load rating (kN)	[C_u]	53.000
Values for approximated geometry:		
Basic dynamic load rating (kN)	[C_{theo}]	402.038
Basic static load rating (kN)	[C_{0theo}]	490.045

Label in the model		HSSFBRROT
Bearing type		SKF NU 320 ECM
Bearing type		Cylindrical roller bearing (single row)
		SKF Explorer
Bearing position (mm)	[y_{lokal}]	40.500
Bearing position (mm)	[y_{global}]	40.500
Attachment of external ring		Free bearing
Inner diameter (mm)	[d]	100.000
External diameter (mm)	[D]	215.000
Width (mm)	[b]	47.000
Corner radius (mm)	[r]	3.000
Number of rolling bodies	[Z]	7
Rolling body reference circle (mm)	[D_{pw}]	152.239
Diameter rolling body (mm)	[D_w]	42.653
Rolling body length (mm)	[L_{we}]	46.527
Diameter, external race (mm)	[d_o]	194.926
Diameter, internal race (mm)	[d_i]	109.553

Calculation with approximate bearings internal geometry (*)

Bearing clearance DIN 620:1988 C0 (67.50 µm)

Basic static load rating (kN)	[C_0]	440.000
Basic dynamic load rating (kN)	[C]	450.000
Fatigue load rating (kN)	[C_u]	51.000

Values for approximated geometry:

Basic dynamic load rating (kN)	[C_{theo}]	450.179
Basic static load rating (kN)	[C_{0theo}]	439.980

Results

Note: the maximum deflection and torsion of the shaft under torque,
the life modification factor aISO, and the bearing's thinnest lubricant film thickness EHL, are predefined for the first load bin.

Shaft

Maximum deflection (µm)	522.407
Position of the maximum (mm)	1081.000
Mass center of gravity (mm)	491.206
Total axial load (N)	0.000
Torsion under torque (°)	-0.563

Bearing

Probability of failure	[n]	10.00	%
Axial clearance	[u_A]	10.00	µm
Lubricant	Oil: Castrol Optigear Synthetic X 320		

Lubricant with additive, effect on bearing lifetime confirmed in tests.
Oil lubrication, on-line filtration, ISO4406 -/17/14

Lubricant - service temperature	[T_B]	65.00	°C
Limit for factor aISO	[a_{ISOmax}]	50.00	
Oil level	[h_{oil}]	0.00	mm

Oil injection lubrication

Rolling bearing service life according to ISO/TS 16281:2008

Shaft 'HSSF' Rolling bearing 'HSSFBRGEN1'

Position (Y-coordinate)	[y]	737.00	mm
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	50.000	
Nominal bearing service life	[L_{nh}]	101567.73	h
Modified bearing service life	[L_{nmh}]	940942.09	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	8.66	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	0.000	µm
Reference rating service life	[L_{nrh}]	201295.34	h
Modified reference rating service life	[L_{nrmh}]	> 1000000	h

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	7.667	5.442	-18.330	19.869	-309.690	0.000	-129.651	335.734
2	4.756	3.331	-11.208	12.175	-195.906	0.000	-83.151	212.822
3	2.871	1.970	-6.618	7.214	-118.781	0.000	-51.501	129.466
4	-0.007	0.071	0.275	0.275	5.185	0.000	0.133	5.187
5	2.886	2.112	7.170	7.729	128.625	0.000	-51.726	138.636

6	5.658	4.113	13.916	15.023	240.515	0.000	-97.784	259.633
7	8.590	6.231	21.059	22.744	352.411	0.000	-143.861	380.643
8	12.462	9.021	30.460	32.911	493.063	0.000	-201.809	532.764
9	15.446	11.179	37.724	40.763	597.850	0.000	-245.071	646.130
10	18.449	13.348	45.017	48.651	700.815	0.000	-287.275	757.409
11	21.467	15.537	52.363	56.593	803.382	0.000	-329.808	868.445

	Displacement of bearing				**Misalignment of bearing**			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	-16.1070	26.1111	38.5714	41.7994	-0.634	1.070	-0.260	0.685
2	-11.5365	19.2368	27.1413	29.4914	-0.412	0.667	-0.174	0.447
3	-8.3018	14.6245	18.9825	20.7185	-0.265	0.403	-0.117	0.290
4	0.2935	7.5042	-4.1892	4.1994	0.053	-0.000	0.010	0.054
5	-8.1118	14.7782	-19.8180	21.4139	0.273	-0.403	-0.115	0.297
6	-12.8302	21.5003	-31.4220	33.9405	0.488	-0.789	-0.198	0.526
7	-17.3807	28.3441	-42.4873	45.9049	0.708	-1.193	-0.286	0.764
8	-23.0167	37.0114	-56.0532	60.5948	0.991	-1.719	-0.404	1.070
9	-27.1893	43.5653	-66.0627	71.4390	1.207	-2.123	-0.493	1.304
10	-31.2347	50.0060	-75.8345	82.0151	1.422	-2.526	-0.583	1.536
11	-35.2594	56.3504	-85.4363	92.4261	1.636	-2.930	-0.672	1.769

Shaft 'HSSF' Rolling bearing 'HSSFBRGEN2'

Position (Y-coordinate)	[y]	789.00	mm
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	50.000	
Nominal bearing service life	[L_{nh}]	144488.09	h
Modified bearing service life	[L_{nmh}]	999719.81	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	9.91	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	0.000	µm
Reference rating service life	[L_{nrh}]	365670.16	h
Modified reference rating service life	[L_{nrmh}]	> 1000000	h

	Bearing reaction force				**Bearing reaction moment**			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	7.110	-5.437	-16.688	18.139	357.337	0.000	151.957	388.305
2	4.446	-3.327	-10.391	11.302	215.924	0.000	92.363	234.849
3	2.697	-1.966	-6.235	6.794	126.515	0.000	54.750	137.854
4	0.007	-0.071	0.267	0.267	-5.178	0.000	0.135	5.180
5	2.682	-2.108	6.768	7.280	-137.390	0.000	54.535	147.817
6	5.241	-4.109	12.851	13.878	-269.767	0.000	110.063	291.355
7	7.885	-6.226	19.130	20.691	-412.624	0.000	169.888	446.229
8	11.290	-9.016	27.245	29.492	-604.477	0.000	250.096	654.171
9	13.887	-11.173	33.416	36.187	-754.018	0.000	312.670	816.276
10	16.466	-13.342	39.561	42.851	-905.950	0.000	376.254	980.975
11	19.032	-15.531	45.657	49.465	-1058.796	0.000	440.073	1146.610

	Displacement of bearing				**Misalignment of bearing**			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	-1.8198	13.9764	3.7501	4.1684	-0.639	1.700	-0.262	0.691
2	-2.0303	7.1699	4.5554	4.9874	-0.416	1.059	-0.175	0.451
3	-1.9083	2.6013	4.4720	4.8622	-0.268	0.641	-0.118	0.293
4	-0.2408	-4.4581	-1.4411	1.4611	0.052	-0.000	0.010	0.053
5	-1.8310	2.7505	-4.9166	5.2465	0.274	-0.641	-0.116	0.297

6	-1.9659	9.4083	-4.7271	5.1196	0.490	-1.254	-0.199	0.529
7	-1.6346	16.1840	-3.6751	4.0222	0.711	-1.895	-0.288	0.767
8	-0.8229	24.7617	-1.6620	1.8545	0.995	-2.732	-0.406	1.075
9	-0.0676	31.2462	0.1879	0.1997	1.211	-3.373	-0.496	1.309
10	0.8006	37.6172	2.2308	2.3701	1.427	-4.014	-0.585	1.542
11	1.6765	43.8912	4.4082	4.7163	1.641	-4.655	-0.674	1.775

Shaft 'HSSF' Rolling bearing 'HSSFBRROT'

Position (Y-coordinate)	[y]	40.50	mm
Life modification factor for reliability[a_1]		1.000	
Life modification factor	[a_{ISO}]	50.000	
Nominal bearing service life	[L_{nh}]	260962.42	h
Modified bearing service life	[L_{nmh}]	999969.73	h
Operating viscosity	[ν]	107.50	mm²/s
Minimum EHL lubricant film thickness	[h_{min}]	0.000	µm
Static safety factor	[S_0]	9.60	
Calculation with approximate bearings internal geometry			
Operating bearing clearance	[Pd]	67.500	µm
Reference rating service life	[L_{nrh}]	479632.19	h
Modified reference rating service life	[L_{nrmh}]	> 1000000	h

	Bearing reaction force				Bearing reaction moment			
	Fx (kN)	Fy (kN)	Fz (kN)	Fr (kN)	Mx (Nm)	My (Nm)	Mz (Nm)	Mr (Nm)
1	6.405	0.000	-15.081	16.384	-35.891	0.000	-15.499	39.095
2	3.993	0.000	-9.282	10.104	-15.231	0.000	-6.427	16.531
3	2.418	0.000	-5.494	6.003	-5.645	0.000	-2.377	6.125
4	0.000	0.000	0.326	0.326	-0.060	0.000	-0.003	0.060
5	2.418	0.000	6.147	6.605	6.574	0.000	-2.482	7.027
6	4.727	0.000	11.700	12.619	22.222	0.000	-9.029	23.986
7	7.138	0.000	17.496	18.896	44.957	0.000	-19.029	48.818
8	10.278	0.000	25.045	27.072	82.559	0.000	-35.846	90.005
9	12.685	0.000	30.829	33.336	116.265	0.000	-50.773	126.868
10	15.088	0.000	36.608	39.595	152.343	0.000	-66.764	166.330
11	17.492	0.000	42.384	45.851	190.279	0.000	-83.830	207.927

	Displacement of bearing				Misalignment of bearing			
	ux (µm)	uy (µm)	uz (µm)	rr (µm)	rx (mrad)	ry (mrad)	rz (mrad)	rr (mrad)
1	-22.3202	186.3056	61.8956	65.7971	0.400	0.000	0.173	0.435
2	-22.9553	179.4314	52.9101	57.6752	0.232	0.000	0.098	0.252
3	-23.5159	174.8191	45.9159	51.5874	0.123	0.000	0.047	0.132
4	7.4680	167.6992	-38.8860	39.5966	0.045	-0.000	0.010	0.046
5	-21.8695	174.9728	-47.8908	52.6480	-0.131	-0.000	0.049	0.140
6	-21.3768	181.6949	-57.4310	61.2804	-0.291	-0.000	0.122	0.316
7	-22.2643	188.5386	-64.8991	68.6119	-0.459	-0.000	0.195	0.499
8	-24.9014	197.2059	-72.2829	76.4519	-0.680	-0.000	0.288	0.738
9	-26.8946	203.7597	-77.2935	81.8389	-0.848	-0.000	0.358	0.920
10	-28.8633	210.2005	-82.0290	86.9589	-1.016	-0.000	0.427	1.102
11	-30.7455	216.5448	-86.5642	91.8622	-1.184	-0.000	0.497	1.284

(*) Note about roller bearings with an approximated bearing geometry:
The internal geometry of these bearings has not been input in the database.
The geometry is back-calculated as specified in ISO 281, from C and C0 (details in the manufacturer's catalog).
For this reason, the geometry may be different from the actual geometry.
This can lead to differences in the service life calculation and, more importantly, the roller bearing stiffness.

Damage (%) [Lreq] (175200.000)

Bin no	B1	B2	B3
1	0.00	0.00	0.00
2	0.00	0.00	0.00
3	0.00	0.00	0.00
4	0.00	0.00	0.00
5	0.01	0.01	0.01
6	0.05	0.03	0.03
7	0.24	0.10	0.11
8	3.51	1.36	1.19
9	3.92	1.48	1.13
10	0.03	0.01	0.01
11	0.00	0.00	0.00
Σ	7.76	3.00	2.48

Utilization (%) [Lreq] (175200.000)

B1	B2	B3
48.17	48.17	48.17

Note: Utilization = (Lreq/Lh)^(1/k)
Ball bearing: k = 3, roller bearing: k = 10/3

B1: HSSFBRGEN1
B2: HSSFBRGEN2
B3: HSSFBRROT

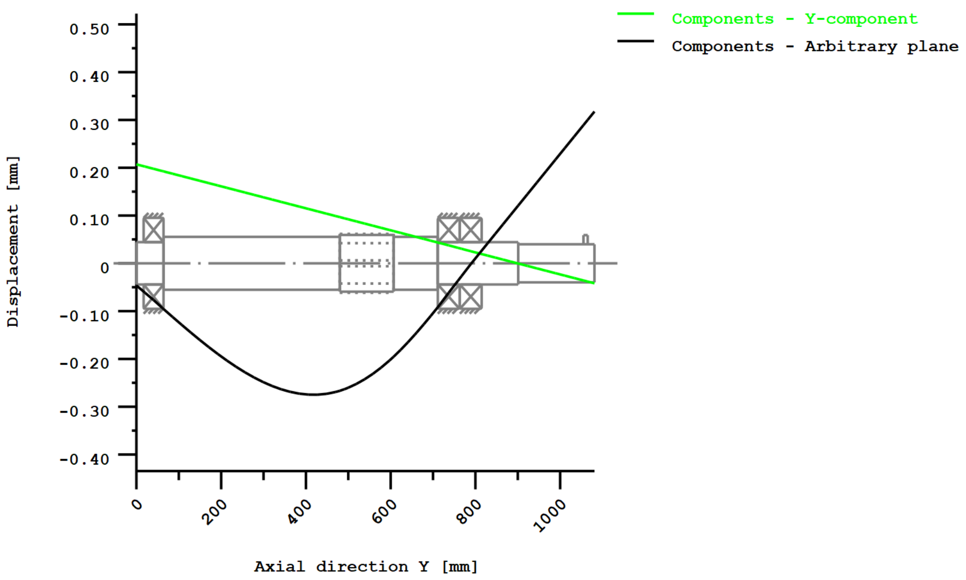
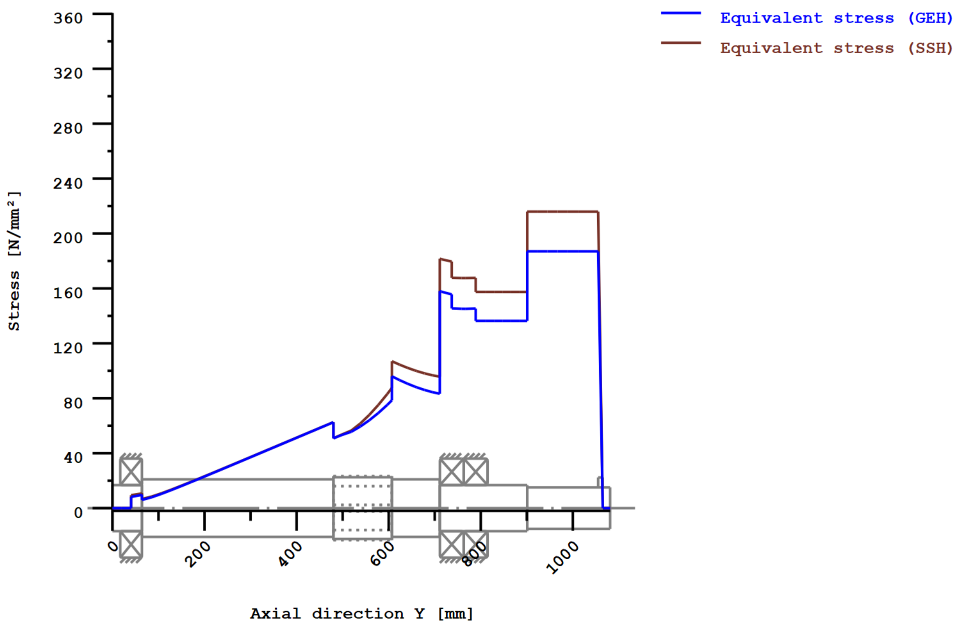


Figure: Deformation (bending etc.) (Arbitrary plane 67.79409544 121)



Nominal stresses, without taking into account stress concentrations
GEH(von Mises): sigV = ((sigB+sigZ,D)^2 + 3*(tauT+tauS)^2)^1/2
SSH(Tresca): sigV = ((sigB-sigZ,D)^2 + 4*(tauT+tauS)^2)^1/2

Figure: Equivalent stress

Strength calculation as specified in the FKM Guideline (6th Edition, 2012)

Summary

HSSF

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

Calculation of service strength and static strength

S-N curve (Woehler line) according Miner elementary

Rolled steel, case-hardening steel

Overload case F1 (chapter 2.4.2): Constant mean stress

Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 1.5	[jG]	1.00

Cross section	Pos (Y co-ord) (mm)	
A-A	661.25	Interference fit
B-B	657.00	Shoulder
C-C	470.00	Shoulder
D-D	597.00	Shoulder

Results:

Cross section	Kfb	KRs	ALGmax	SD	SS	SB
A-A	2.72	1.00	0.57	2.35	3.27	4.62
B-B	1.07	0.88	0.27	5.09	3.27	4.62
C-C	1.74	0.88	0.63	2.13	4.83	6.81
D-D	1.29	0.88	0.36	3.71	4.21	5.94
Nominal safety:				1.35	1.40	1.85

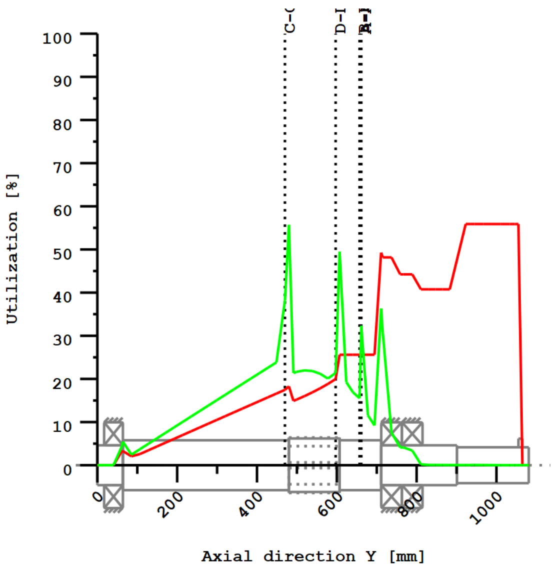
Abbreviations:
Kfb: Notch factor bending
KRs: Surface factor
ALGmax: Highest utilization
SD: Safety endurance limit
SS: Safety against yield point
SB: Safety against tensile stress

Service life and damage

System service life (h)	[Hatt]	1000000.00
Damage to system (%)	[D]	0.00

Utilization (%) [Smin/S]

Cross section	Static	Endurance
A-A	42.750	57.403
B-B	42.750	26.508
C-C	29.014	63.445
D-D	33.253	36.400
Maximum utilization (%)	[A]	63.445



Utilization = Smin/S (%)

Figure: Strength

Calculation details

General statements

Label	HSSF	
Drawing		
Length (mm)	[l]	1081.00
Speed (1/min)	[n]	1495.38

Material	18CrNiMo7-6
Material type	Case-carburized steel
Material treatment	case-hardened
Surface treatment	No

	Tension/Compression	Bending	Torsion	Shearing
Load factor static calculation	1.700	1.700	1.700	1.700
Load factor endurance limit	1.000	1.000	1.000	1.000

Rolled steel, case-hardening steel

Base stress according FKM chapter 5.1:

Tensile strength (N/mm²)	[Rm,N]	1200.00
Yield point (N/mm²)	[Rp,N]	850.00
Fatigue limit (N/mm²)	[σzdWN]	480.00
Fatigue limit (N/mm²)	[σbWN]	510.00

Fatigue limit (N/mm²)	[τtWN]	305.00		
Fatigue limit (N/mm²)	[τsWN]	280.00		
Breaking elongation (%)	[A]	8.00		
Reference diameter (mm)	[deffNm, deffNp]	16.00	16.00	
Required life time	[H]	175200.00		
Number of load cycles	[NL]	15719483077		
Service strength for a load spectrum				
S-N curve (Woehler lines) according to Miner elementary according to FKM guideline				
Temperature (°C)	[Temperatur]	20.000		
Temperature duration (h)	[TemperaturD]	175200.000		
Temperature coefficients	[KTm, KTp, KTD]	1.000	1.000	1.000
	[KTtm, KTtp]	1.000	1.000	
Internal stress coefficient	[KEs, KEt]	1.000	1.000	
Additional coefficients	[KA, KW, KfW]	1.000	1.000	1.000
	[KNL, KNLE]	1.000	1.000	
Protective layer factor	[KS]	1.000		

Material properties:

[fσZ, fσD, fτ, Rpmax]	1.000	1.000	0.577	1150.0
[fWt, fWs]	0.577	0.400		
[aM, bM, aTD]	0.35000	-0.100	1.400	
[aG, bG, aRsig, RmNmin]	0.500	2700.0	0.220	400.0
[MS, MT]	0.1959	0.1131		
[kσ, kτ]	15	25		
[kDσ, kDτ]	0	0		
[NDσ, NDτ]	1e+006	1e+006		
[NDσII, NDτII]	0	0		

Thickness of raw material (mm)	[d.eff]	140.00	
Material data calculated acc. FKM directive with Kdm, Kdp			
Geometric size factors (Kdm, Kdp) calculated from raw diameter			
Material strength calculated from size of raw material			
Constants	[adm, adp]	0.370	0.370
Size factors	[Kdm, Kdp]	0.704	0.704
Tensile strength (N/mm²)	[Rm]	845.34	
Yield point (N/mm²)	[Rp]	598.79	
σzdW (N/mm²)	[σzdW]	338.14	
σbW (N/mm²)	[σbW]	359.27	
τtW (N/mm²)	[τtW]	214.86	
τsW (N/mm²)	[τsW]	197.25	

Overload case F1 (chapter 2.4.2): Constant mean stress		
Safety number according Chapter 1.5	[jm]	1.85
Safety number according Chapter 1.5	[jp]	1.40
Safety number according Chapter 1.5	[jmt]	1.40
Safety number according Chapter 1.5	[jpt]	1.00
Safety number according Chapter 2.5	[jF]	1.35
Safety number according Chapter 1.5	[jG]	1.00

Cross section 'A-A' Interference fit

Comment	Fit at bearing 1	
Position (Y-Coordinate) (mm)	[y]	661.250

External diameter (mm)	[da]	125.000
Inner diameter (mm)	[di]	0.000
Notch effect		Interference fit
Characteristics:	Slight interference fit	
Mean roughness (µm)	[Rz]	8.000

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value	[Fzdm, Mbm, Tm, Fqm]	6.4	0.0	-25807.8	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	10284.6	0.0	105826.5
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	10.9	17483.9	43873.3	179905.1
Cross section, moment of resistance: (mm²)					
[A, Wb, Wt, A]		12271.8	191747.6	383495.2	12271.8

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	4.681	3812.882	9426.818	38237.732
2	1.0330e-003	4.292	2387.169	5872.444	23706.962
3	1.0657e-002	4.031	1449.002	3554.374	14237.042
4	9.3936e+000	0.045	8.227	-0.000	292.881
5	2.7248e+001	4.039	1463.037	-3554.374	14778.100
6	1.9304e+001	4.420	2835.903	-6954.210	28669.671
7	1.3095e+001	4.808	4257.278	-10508.584	43203.825
8	2.3136e+001	5.299	6095.956	-15144.724	62171.997
9	7.7924e+000	5.670	7497.293	-18699.098	76719.182
10	1.8943e-002	6.035	8892.786	-22253.472	91271.615
11	5.5000e-005	6.394	10284.626	-25807.846	105826.520

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	0.001	0.000	-67.296	0.000
[σaz, σab, τat, τas]	0.000	53.636	0.000	11.498
[σzmax, σbmax, τtmax, τsmax]	0.001	91.182	114.404	19.547

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Notch effect coefficient	[β(dB)]	2.591	2.591	1.716	1.358
[dB] (mm) 40.0, [rB] (mm) 2.4, [r] (mm) 7.5					
Support number	[n(r)]	1.097	1.097	1.076	1.076
Support number	[n(rB)]	1.163	1.163	1.135	1.135
Support number	[n(d)]	1.008	1.008	1.011	1.011
Mechanical material support factor	[nwm]	1.052	1.052	1.052	1.052
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	2.745	2.724	1.790	1.432
Roughness factor	[KR]	1.000	1.000	1.000	1.000
Roughness factor is included into the notch effect coefficient					
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.200
Design coefficient	[KWK]	2.287	2.270	1.492	1.193
Fatigue limit of part (N/mm²)	[SWK]	147.821	148.971	132.222	165.327

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.846	0.847	0.942	0.954
Permissible amplitude (N/mm²)	[SAK]	124.990	126.141	124.612	157.717
Effective Miner sum	[DM]	0.829	0.335	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SBK]	124.990	126.141	124.612	157.717
Rate of utilization	[aBK]	0.000	0.574	0.000	0.098

Calculation of the combined stress types:
Rate of utilization for the combined load components
a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	116.561
Rate of utilization	[aBKv_1]	0.574

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	116.561
Rate of utilization	[aBKv_2]	0.098

Highest utilization	[aBKmax]	0.574
Safety endurance limit assessment	[S.Dauer]	2.352
Required safety	[jD]	1.350
Result (%)	[S/jD]	174.2

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.700		1.330
Plastic support number	[npl]	1.0000	1.3858	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	598.79	829.82	459.79	345.71
Rate of utilization	[aSK]	0.000	0.154	0.348	0.079

Rate of utilization for the combined load components:
a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	218.126
Rate of utilization	[aSKvBn]	0.381

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	232.009
Rate of utilization	[aSKvQn]	0.427

Highest utilization	[aSKmax]	0.427

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	4.623
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	249.9
Safety against yield point	[S.Rp]	3.275
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	233.9

Cross section 'B-B' Shoulder

Comment	Y= 657.00mm	
Position (Y-Coordinate) (mm)	[y]	657.000
External diameter (mm)	[da]	125.000

Inner diameter (mm)	[di]	0.000	
Notch effect		Shoulder	
[D, r, t] (mm)	126.000	2.000	0.500
Mean roughness (µm)	[Rz]	8.000	

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value	[Fzdm, Mbm, Tm, Fqm]	6.4	0.0	-25807.8	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	10734.4	0.0	105822.8
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	10.9	18248.4	43873.3	179898.8
Cross section, moment of resistance: (mm²)					
	[A, Wb, Wt, A]	12271.8	191747.6	383495.2	12271.8

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	4.681	3975.400	9426.818	38241.426
2	1.0330e-003	4.292	2487.930	5872.444	23710.653
3	1.0657e-002	4.031	1509.516	3554.374	14240.728
4	9.3936e+000	0.045	9.463	-0.000	288.876
5	2.7248e+001	4.039	1525.835	-3554.374	14774.390
6	1.9304e+001	4.420	2957.740	-6954.210	28665.966
7	1.3095e+001	4.808	4440.886	-10508.584	43200.123
8	2.3136e+001	5.299	6360.178	-15144.724	62168.296
9	7.7924e+000	5.670	7823.341	-18699.098	76715.481
10	1.8943e-002	6.035	9280.683	-22253.472	91267.914
11	5.5000e-005	6.394	10734.381	-25807.846	105822.819

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	0.001	0.000	-67.296	0.000
[σaz, σab, τat, τas]	0.000	55.982	0.000	11.498
[σzmax, σbmax, τtmax, τsmax]	0.001	95.169	114.404	19.546

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	1.620	1.255	1.182	1.091
References stress slope	[G']	1.438	1.438	0.575	0.575
Support number	[n(r)]	1.168	1.168	1.158	1.158
Support number	[n(d)]	1.008	1.008	1.011	1.011
Mechanical material support factor	[nwm]	1.052	1.052	1.052	1.052
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	1.387	1.066	1.010	1.000
Roughness factor	[KR]	0.876	0.876	0.928	0.928
Surface stabilization factor	[KV]	1.200	1.100	1.100	1.100
Design coefficient	[KWK]	1.274	1.098	0.989	0.979
Fatigue limit of part (N/mm²)	[SWK]	265.416	307.933	199.518	201.391
Calculation with principal mean stress:					
Mean stress coefficient	[KAK]	0.914	0.926	0.962	0.962
Permissible amplitude (N/mm²)	[SAK]	242.585	285.103	191.908	193.781
Effective Miner sum	[DM]	0.829	0.335	0.3	0.3

Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SBK]	242.585	285.103	191.908	193.781
Rate of utilization	[aBK]	0.000	0.265	0.000	0.080

Calculation of the combined stress types:
Rate of utilization for the combined load components
a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	116.561
Rate of utilization	[aBKv_1]	0.265

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	116.561
Rate of utilization	[aBKv_2]	0.080

Highest utilization	[aBKmax]	0.265
Safety endurance limit assessment	[S.Dauer]	5.093
Required safety	[jD]	1.350
Result (%)	[S/jD]	377.2

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]	1.700		1.330	
Plastic support number	[npl]	1.0000	1.3858	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	598.79	829.82	459.79	345.71
Rate of utilization	[aSK]	0.000	0.161	0.348	0.079

Rate of utilization for the combined load components:
a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	219.823
Rate of utilization	[aSKvBn]	0.384

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	232.008
Rate of utilization	[aSKvQn]	0.427

Highest utilization [aSKmax] 0.427

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	4.623
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	249.9
Safety against yield point	[S.Rp]	3.275
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	233.9

Cross section 'C-C' Shoulder

Comment Y= 470.00mm

Position (Y-Coordinate) (mm)		[y]	470.000
External diameter (mm)		[da]	125.000
Inner diameter (mm)		[di]	0.000
Notch effect			Shoulder
[D, r, t] (mm)	138.000	5.000	6.500

Mean roughness (µm) [Rz] 8.000

		Tension/Compression	Bending	Torsion	Shearing
Load: (N) (Nm)					
Mean value	[Fzdm, Mbm, Tm, Fqm]	0.0	0.0	-0.1	0.0
Amplitude	[Fzda, Mba, Ta, Fqa]	0.0	19397.8	0.0	45462.3
Maximum value	[Fzdmax, Mbmax, Tmax, Fqmax]	0.1	32976.2	0.1	77285.9
Cross section, moment of resistance: (mm²)					
	[A, Wb, Wt, A]	12271.8	191747.6	383495.2	12271.8

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	0.000	7085.093	0.028	16772.901
2	1.0330e-003	0.000	4410.338	0.017	10492.730
3	1.0657e-002	0.000	2658.847	0.011	6390.489
4	9.3936e+000	0.045	45.691	-0.000	94.774
5	2.7248e+001	0.000	2742.096	-0.011	6215.138
6	1.9304e+001	0.000	5308.281	-0.021	12229.509
7	1.3095e+001	-0.000	7979.343	-0.031	18506.378
8	2.3136e+001	0.000	11450.070	-0.045	26683.160
9	7.7924e+000	0.000	14103.586	-0.056	32947.161
10	1.8943e-002	-0.000	16752.262	-0.066	39205.939
11	5.5000e-005	-0.000	19397.757	-0.077	45462.267

Stresses: (N/mm²)				
[σmz, σmb, τmt, τms]	0.000	0.000	-0.000	0.000
[σaz, σab, τat, τas]	0.000	101.163	0.000	4.939
[σzmax, σbmax, τtmax, τsmax]	0.000	171.977	0.000	8.397

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	2.116	1.953	1.451	1.225
References stress slope	[G']	0.530	0.530	0.230	0.230
Support number	[n(r)]	1.112	1.112	1.100	1.100
Support number	[n(d)]	1.008	1.008	1.011	1.011
Mechanical material support factor	[nwm]	1.052	1.052	1.052	1.052
The support factor is determined with the support factor as defined by Stieler.					
Notch effect coefficient beta	[Kf]	1.903	1.743	1.305	1.114
Roughness factor	[KR]	0.876	0.876	0.928	0.928
Surface stabilization factor	[KV]	1.200	1.200	1.200	1.100
Design coefficient	[KWK]	1.704	1.571	1.152	1.083
Fatigue limit of part (N/mm²)	[SWK]	198.457	215.259	171.230	182.135
Calculation with principal mean stress:					
Mean stress coefficient	[KAK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SAK]	198.457	215.259	171.230	182.135
Effective Miner sum	[DM]	1	0.332	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SBK]	198.457	215.259	171.230	182.135
Rate of utilization	[aBK]	0.000	0.634	0.000	0.037

Calculation of the combined stress types:
Rate of utilization for the combined load components
a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	0.000
Rate of utilization	[aBKv_1]	0.634

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	0.000
Rate of utilization	[aBKv_2]	0.037

Highest utilization	[aBKmax]	0.634
Safety endurance limit assessment	[S.Dauer]	2.128
Required safety	[jD]	1.350
Result (%)	[S/jD]	157.6

STATIC STRENGTH ASSESSMENT:
Total safety factor according chapter 1.5.3 [jges] 1.400
(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]	1.700		1.330	
Plastic support number	[npl]	1.0000	1.3858	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	598.79	829.82	459.79	345.71
Rate of utilization	[aSK]	0.000	0.290	0.000	0.034

Rate of utilization for the combined load components:
a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	171.977
Rate of utilization	[aSKvBn]	0.290

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	14.545
Rate of utilization	[aSKvQn]	0.034

Highest utilization [aSKmax] 0.290

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	6.812
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	368.2

Safety against yield point	[S.Rp]	4.825
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	344.7

Cross section 'D-D' Shoulder

Comment Y= 597.00mm

Position (Y-Coordinate) (mm)		[y]	597.000
External diameter (mm)		[da]	134.000
Inner diameter (mm)		[di]	0.000

Notch effect Shoulder
[D, r, t] (mm) 138.000 5.000 2.000
Mean roughness (µm) [Rz] 8.000

Tension/Compression Bending Torsion Shearing

Load: (N) (Nm)

Mean value [Fzdm, Mbm, Tm, Fqm]	6.4	0.0	-23775.7	0.0
Amplitude [Fzda, Mba, Ta, Fqa]	0.0	17022.7	0.0	93869.3
Maximum value [Fzdmax, Mbmax, Tmax, Fqmax]	10.9	28938.5	40418.8	159577.9

Cross section, moment of resistance: (mm²)

[A, Wb, Wt, A]	14102.6	236218.7	472437.4	14102.6

Load spectrum, load base values (Mean-value + Amplitude):

No.	Frequency (%)	Tens./Compres. (N)	Bending (Nm)	Torsion (Nm)	Shearing (N)
1	1.2440e-003	4.681	6249.721	8684.551	33948.209
2	1.0330e-003	4.292	3898.597	5410.048	21056.324
3	1.0657e-002	4.031	2357.327	3274.503	12655.208
4	9.3936e+000	0.045	25.092	-0.000	230.920
5	2.7248e+001	4.039	2402.522	-3274.503	13081.841
6	1.9304e+001	4.420	4660.087	-6406.636	25405.814
7	1.3095e+001	4.808	7007.089	-9681.139	38301.075
8	2.3136e+001	5.299	10053.784	-13952.230	55131.543
9	7.7924e+000	5.670	12381.583	-17226.732	68039.817
10	1.8943e-002	6.035	14703.876	-20501.235	80953.339
11	5.5000e-005	6.394	17022.674	-23775.738	93869.330

Stresses: (N/mm²)

[σmz, σmb, τmt, τms]	0.000	0.000	-50.326	0.000
[σaz, σab, τat, τas]	0.000	72.063	0.000	8.875
[σzmax, σbmax, τtmax, τsmax]	0.001	122.507	85.554	15.087

FATIQUE PROOF:

Total safety factor according chapter 2.5.3 [jD] 1.350

(Formula: jD = jF*jG/KTD)

		Tension/Compression	Bending	Torsion	Shearing
Stress concentration factor	[a]	1.735	1.443	1.248	1.124
References stress slope	[G']	0.562	0.562	0.230	0.230
Support number	[n(r)]	1.115	1.115	1.100	1.100
Support number	[n(d)]	1.007	1.007	1.010	1.010
Mechanical material support factor	[nwm]	1.052	1.052	1.052	1.052

The support factor is determined with the support factor as defined by Stieler.

Notch effect coefficient beta	[Kf]	1.556	1.285	1.124	1.022
Roughness factor	[KR]	0.876	0.876	0.928	0.928
Surface stabilization factor	[KV]	1.200	1.200	1.100	1.100
Design coefficient	[KWK]	1.415	1.189	1.092	0.999
Fatigue limit of part (N/mm²)	[SWK]	239.014	284.339	180.631	197.365

Calculation with principal mean stress:

Mean stress coefficient	[KAK]	0.929	0.940	0.968	0.971
Permissible amplitude (N/mm²)	[SAK]	221.941	267.266	174.940	191.674
Effective Miner sum	[DM]	0.829	0.333	0.3	0.3
Coefficient service strength	[KBK]	1.000	1.000	1.000	1.000
Permissible amplitude (N/mm²)	[SBK]	221.941	267.266	174.940	191.674
Rate of utilization	[aBK]	0.000	0.364	0.000	0.063

Calculation of the combined stress types:

Rate of utilization for the combined load components

a)For outer surface (shear stress = 0)

Equivalent mean stress (N/mm²)	[SmV_1]	87.167
Rate of utilization	[aBKv_1]	0.364

b) For neutral line (Bending stress = 0)

Equivalent mean stress (N/mm²)	[SmV_2]	87.167
Rate of utilization	[aBKv_2]	0.063

Highest utilization	[aBKmax]	0.364
Safety endurance limit assessment	[S.Dauer]	3.709
Required safety	[jD]	1.350
Result (%)	[S/jD]	274.7

STATIC STRENGTH ASSESSMENT:

Total safety factor according chapter 1.5.3 [jges] 1.400

(Formula: jges = jG*Max(jm/KTm*Rp/Rm, jp/KTp, jmt/KTtm*Rp/Rm, jpt/KTtp))

		Tension/Compression	Bending	Torsion	Shearing
Plastic notch factor	[Kpb, Kpt]		1.700	1.330	
Plastic support number	[npl]	1.0000	1.3858	1.3300	1.0000
Strength of part (N/mm²)	[SSK]	598.79	829.82	459.79	345.71
Rate of utilization	[aSK]	0.000	0.207	0.260	0.061

Rate of utilization for the combined load components:

a)For outer surface (shear stress = 0)

Equivalent stress (N/mm²)	[SvBn]	192.267
Rate of utilization	[aSKvBn]	0.333

b) For neutral line (Bending stress = 0)

Equivalent stress (N/mm²)	[SvQn]	174.315
Rate of utilization	[aSKvQn]	0.322

Highest utilization [aSKmax] 0.333

Safety for fracture and yield stresses:

Safety against fracture	[S.Rm]	5.944
Required safety	[jm/Ktm]	1.850
Result (%)	[S/jm]	321.3

Safety against yield point	[S.Rp]	4.210
Required safety	[jp/KTp]	1.400
Result (%)	[S/jp]	300.7

Important remarks concerning strength calculation according to FKM-Guideline:

- Calculation with nominal stresses
- Regulation for proof: Utilization <= 1
- Currently the following restrictions still apply::
  Only for axially symmetrical shafts
- Assumption for calculating the notch factor for shearing:
  ßS = 1.0 + (ßT - 1.0) / 2.0 (according to Prof. Haibach)
- Thread: Determination of notch factor as circumferential groove
- Slight interference fit: determination of the notch factor according to fig. 5.3.11 b) with p = 20MPa
- Proven safety: Effective safety according to special formula,
  condition: safety > required safety or result > 100%

End of Report lines: 1108

A.3 KISSsoft Report - Thermal analysis

A.3.1 KISSsoft Report - High speed shaft

———————————— KISSsoft Release 03/2017 A ————————————

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———————————— File ————————————

Name : Unnamed

Changed by: Joana Mêda de Sousa on: 24.09.2017 at: 15:26:48

THERMALLY SAFE OPERATING SPEED CALCULATION

(according to DIN ISO 15312 and DIN 732)

Lubricant		Oil: Castrol Optigear Synthetic X 320	
Lubrication type:			
Oil-groove lubrication			
Mean bearing temperature	[T_m]	90.000	°C
Temperature of bearing environment	[T_u]	40.000	°C
Lubricant - service temperature	[T_B]	65.000	°C
Lubricant temperature - Reference conditions	[T_{ref}]	70.000	°C

Shaft 'HSSF', Rolling bearing 'HSSFBRGEN1':

Thermal nominal speed according to DIN ISO 15312:

Type of support		Taper roller bearing (single row)	
Bearing number		SKF 30320 J2	
Design series		303	
Speed	[n]	1495.385	1/min
Coefficient	[f_{0r}]	3.000	
(Depends upon type of design and lubrication at reference conditions)			
Coefficient	[f_{1r}]	0.000400	
(Depends upon type of design and load at reference conditions)			
Heat sink reference surface	[A_s]	50964.487	mm²
Reference load	[P_{1r}]	24.500	kN
Bearing mean diameter	[d_m]	157.500	mm
Bearing-specific reference heat flow density	[q_r]	15.896	kW/m²
kinematic viscosity (for reference conditions)	[v_r]	12.000	mm²/s
Thermal nominal speed	[$n_{\theta r}$]	4569.143	1/min

Thermally safe operating speed according to DIN 732:

Coefficient	[f_0]	3.000	
(Depends upon type of design and lubrication)			
Coefficient	[f_1]	0.000400	
(Depends upon type of design and load)			
Temperature difference	[$\Delta\theta=\theta_o-\theta_i$]	5.000	°C
Lubricant Oil-volume	[V_L]	0.500	l/min
Heat flow (dissipated by the lubricant)	[Φ_L]	0.071	kW
Heat flow (dissipated by the bearing support surface)	[Φ_S]	0.810	kW
Total heat flow	[Φ]	0.881	kW
Dynamic equivalent load	[P_1]	27821.069	N
kinematic viscosity at service temperature	[v]	107.500	mm²/s
Lubricant film parameter	[K_L]	3.961	
Charge parameter	[K_P]	0.951	
Speed ratio	[f_n]	0.344	
Thermally safe operating speed	[n_θ]	1572.874	1/min

Shaft 'HSSF', Rolling bearing 'HSSFBRGEN2':

Thermal nominal speed according to DIN ISO 15312:

Type of support	Taper roller bearing (single row)		
Bearing number	SKF 30320 J2		
Design series	303		
Speed	[n]	1495.385	1/min
Coefficient	[f_{0r}]	3.000	
(Depends upon type of design and lubrication at reference conditions)			
Coefficient	[f_{1r}]	0.000400	
(Depends upon type of design and load at reference conditions)			
Heat sink reference surface	[A_s]	50964.487	mm²
Reference load	[P_{1r}]	24.500	kN
Bearing mean diameter	[d_m]	157.500	mm
Bearing-specific reference heat flow density	[q_r]	15.896	kW/m²
kinematic viscosity (for reference conditions)	[v_r]	12.000	mm²/s
Thermal nominal speed	[$n_{\theta r}$]	4569.143	1/min

Thermally safe operating speed according to DIN 732:

Coefficient	[f_0]	3.000	
(Depends upon type of design and lubrication)			
Coefficient	[f_1]	0.000400	
(Depends upon type of design and load)			
Temperature difference	[$\Delta\theta=\theta_o-\theta_i$]	5.000	°C
Lubricant Oil-volume	[V_L]	0.500	l/min
Heat flow (dissipated by the lubricant)	[Φ_L]	0.071	kW
Heat flow (dissipated by the bearing support surface)	[Φ_S]	0.810	kW
Total heat flow	[Φ]	0.881	kW
Dynamic equivalent load	[P_1]	26121.982	N
kinematic viscosity at service temperature	[v]	107.500	mm²/s
Lubricant film parameter	[K_L]	3.961	
Charge parameter	[K_P]	0.893	
Speed ratio	[f_n]	0.352	
Thermally safe operating speed	[n_θ]	1606.339	1/min

Shaft 'HSSF', Rolling bearing 'HSSFBRROT':

Thermal nominal speed according to DIN ISO 15312:

Type of support	Cylindrical roller bearing (single row)		
Bearing number	SKF NU 320 ECJ		
Design series	3		
Speed	[n]	1495.385	1/min
Coefficient	[f_{0r}]	2.000	
(Depends upon type of design and lubrication at reference conditions)			
Coefficient	[f_{1r}]	0.000350	
(Depends upon type of design and load at reference conditions)			
Heat sink reference surface	[A_s]	46511.279	mm²
Reference load	[P_{1r}]	22.000	kN
Bearing mean diameter	[d_m]	157.500	mm
Bearing-specific reference heat flow density	[q_r]	16.000	kW/m²
kinematic viscosity (for reference conditions)	[v_r]	12.000	mm²/s
Thermal nominal speed	[$n_{\theta r}$]	5537.936	1/min

Thermally safe operating speed according to DIN 732:

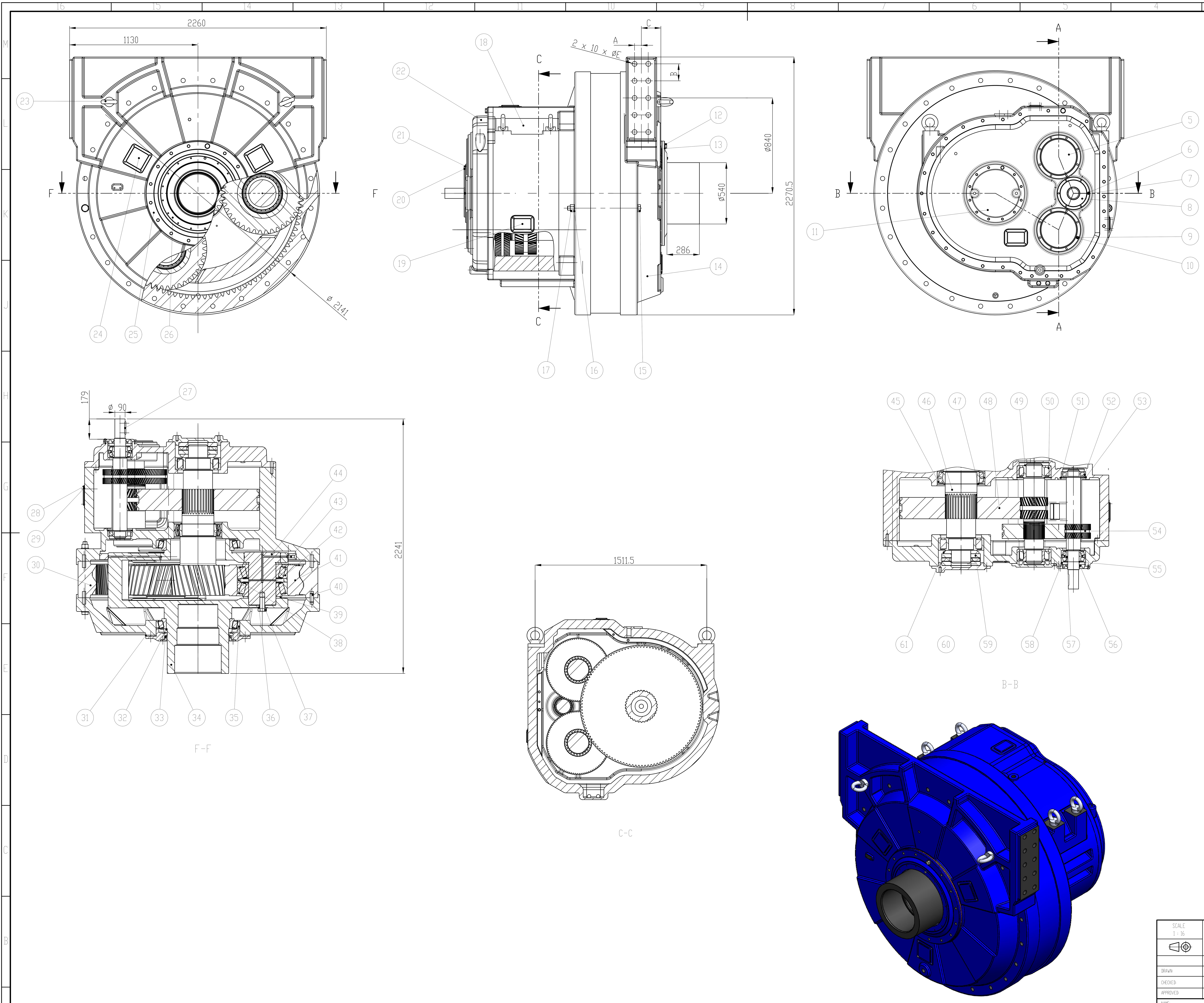
Coefficient	[f_0]	2.000	
(Depends upon type of design and lubrication)			
Coefficient	[f_1]	0.000350	
(Depends upon type of design and load)			
Temperature difference	[$\Delta\theta=\theta_o-\theta_i$]	5.000	°C
Lubricant Oil-volume	[V_L]	0.500	l/min
Heat flow (dissipated by the lubricant)	[Φ_L]	0.071	kW
Heat flow (dissipated by the bearing support surface)	[Φ_S]	0.721	kW
Total heat flow	[Φ]	0.792	kW
Dynamic equivalent load	[P_1]	22860.293	N
kinematic viscosity at service temperature	[v]	107.500	mm²/s
Lubricant film parameter	[K_L]	4.048	
Charge parameter	[K_P]	0.923	
Speed ratio	[f_n]	0.344	
Thermally safe operating speed	[n_θ]	1906.369	1/min

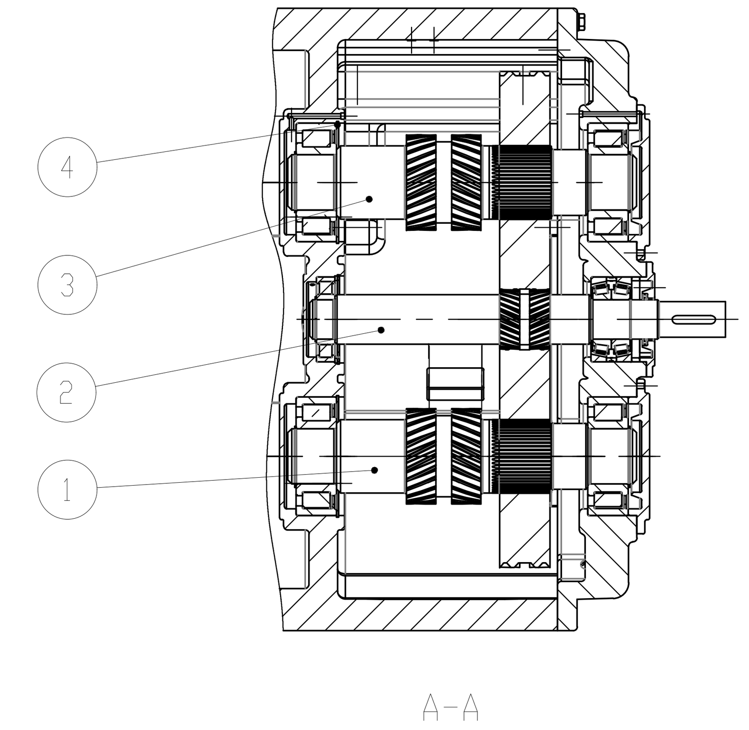
The reference conditions for calculating the thermal nominal speed are taken from the DIN ISO 15312 standard.

End of Report lines: 134

Appendix B

Assembly drawing





ITEM NO.	QTY.	DESCRIPTION	STANDARD
1	1	Inferior intermadiate shaft	
2	1	High speed shaft	
3	1	Superior intermediate shaft	
4	2	Internal retaining ring	DIN 472 - 300 x 5
5	2	External cover - 8 x M12	
6	3	Plain washer	ISO 7089 - Ø10
7	3	Hexagonal bolt	ISO 4017 - M10 x 30 - 10.9
8	1	Output cover - 6 x M10	
9	1	Plain washer	ISO 7089 - Ø12
10	1	Hexagonal bolt	ISO 4017 - M12 x 35 - 10.9
11	1	External cover - 12 x M16	
12	2	Hexagonal bolt	ISO 4017 - M24 x 90 - 10.9
13	2	Plain washer	ISO 7089 - Ø14
14	1	Main housing lid (rot) - 24 x M36	
15	1	Hexagonal bolt	ISO 4014 - M36 x 620 - 10.9
16	2	Plain washer	ISO 7089 - Ø30
17	1	Hexagonal nut	ISO 4035 - M36 - 10.9
18	1	Housing	
19	3	Inspection cover - 4 x M8	
20	1	Hexagonal bolt	ISO 4017 - M16 x 35 - 10.9
21	1	Plain washer	ISO 7089 - Ø6
22	1	Main housing lid (gen) - 29 x M24	
23	6	Lifting eye bolt	DIN 580 - M42 x 3
24	3	Inspection cover - 4 x M10	
25	1	Main input cover - 16 x M24	
26	1	Secondary input cover - 16 x M16	
27	1	Parallel key A	DIN 6885 - 25 x 14 x 110
28	1	Hexagonal bolt	ISO 4017 - M8 x 25 - 10.9
29	1	Plain washer	ISO 7089 - Ø8
30	1	Ring gear	
31	1	Laminated shim - Ø920 x Ø756 x 10	
32	1	V-ring	
33	1	Input shaft seal	
34	1	Planet carrier	
35	2	Tapered roller bearing - Ø558.8 x Ø736.6 x 88.1	
36	1	Spring lock washer W	DIN 127 - Ø42
37	1	Hexagonal bolt	ISO 4017 - M42 x 100 - 10.9
38	3	Spacer sleeve - Ø380 x Ø440 x 31.5	
39	6	Spacer washer - Ø240 x Ø330 x 15	
40	1	Clevis pin (headless)	DIN/EN 22340 - Ø30 x 80
41	3	Planet gear	
42	6	Tapered roller bearing - Ø240 x Ø440 x 127	
43	1	Lubricant distribuition sleeve	
44	3	Planet gear shaft	
45	1	Spherical roller bearing - Ø280 x Ø420 x 106	
46	1	Sun shaft	
47	1	External retaining ring	DIN 471 - 280 x 5
48	1	2nd stage gear	
49	4	External retaining ring	DIN 471 - 140 x 4
50	4	Cylindrical roller bearing - Ø100 x Ø300 x 102	
51	1	Internal retaining ring	DIN 472 - 215 x 5
52	1	Cylindrical roller bearing - Ø100 x Ø215 x 47	
53	1	Spacer washer - Ø197 x Ø220 x 6	
54	2	3rd stage gear	
55	2	External retaining ring	DIN 471 - 100 x 3
56	1	Output shaft seal	
57	1	Tapered roller bearings matched face-to-face - 2 x Ø100 x Ø215 x 51.5	
58	1	Spacer sleeve - Ø195 x Ø215 x 20	
59	1	Cylind. roller thrust bearing - Ø170 x Ø340 x 103	
60	1	Cylindircal roller bearing - Ø220 x Ø400 x 108	
61	1	Spacer sleeves - Ø315 x Ø170 x 30	

SCALE 1 : 16	Tolerances according to: ISO 8015 General tolerances acc. to: ISO 2768 - mH ISO 13920 General roughness acc. to: ISO 1302 General chamfer and fillets acc. to: ISO 13715	MATERIAL :	WEIGHT: 18 ton
	DATE	NAME	
DRAWN:	2017-09-18	Joana Mêda de Sousa	
CHECKED:			
APPROVED:			
NAME: Joana Mêda de Sousa	INSTITUTION DEPARTMENT: MIEM		
INSTITUTION: FEUP Faculdade de Engenharia da Universidade do Porto RuaDr. Roberto Frias s/n 4200-465 Porto, Portugal	TITTLE: Wind turbine gearbox	DWG NO.:	
Rev.:	EDITION: 2017-10-11	LANG: en	A1 1/1